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TEST RESULTS REPORT AND TECHNOLOGY DEVELOPMENT REPORT, HLH/ATC DRIVE SYSTEM AFT ROTOR SHAFT BEARING TEST

Joseph W. Lenski, Jr.

Boeing Vertol Company

Prepared for:

Army Aviation Systems Command

January 1974

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13. ABSTRACT

This report presents the results of a program, conducted between September 1971 and March 1973, to define criteria and to design, fabricate, and test simulated HLH aft rotor shaft tapered roller bearings. Tests were performed in a back-to-back test rig which simulated the expected loading, lubrication, housing stiffness, and preload of the HLH aft rotor shaft application. Technical test inspection and evaluation will be used to determine bearing performance and adequacy of bearing fatigue life under predicted operating conditions. Trade studies conducted of the HLH aft rotor shaft support bearing show that large-pitch-diameter and high-contact-angle duplex tapered roller bearings provide the most effective method of reacting the HLH rotor loads. This bearing configuration resulted in the lightest weight and most compact bearing system for the HLH aft rotor shaft bearing application. The test bearings designed for the HLH rotor shaft have pitch diameters of 23.5 and 24.0 inches and contact angles of 35 and 45 degrees. These bearings provide substantial weight savings and improved life and reliability compared to other conventional rotor shaft support bearing configurations. Endurance testing was conducted for approximately 1100 hours, at which time one bearing was damaged due to fatigue spalling on the outer race and rollers. The remaining three bearings tested showed no damage after operating at 112 percent of equivalent design loads. Analysis of the failed bearing ascribed this failure to concentrated loads induced by the test fixture; specifically, to the damping plate which did not duplicate the aircraft arrangement and the effect of four-point mounting of the housing. Based on the test results obtained, it is calculated that the equivalent aircraft hours for the actual HLH aft rotor shaft bearings will be satisfactory, and no problems are expected during DSTR and prototype aircraft testing.

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	ROLE	WT	ROLE	WT	ROLE	WT
Tapered roller bearings						
Helicopter aft rotor shaft						
Test program						
Large pitch diameter						
High contact angle						
Lightest weight						
Most compact bearing						
Weight savings						
Improved life and reliability						
Endurance testing						
Satisfactory results						

DEPARTMENT OF THE ARMY
U. S. ARMY AIR MOBILITY RESEARCH & DEVELOPMENT LABORATORY
EUSTIS DIRECTORATE
FORT EUSTIS, VIRGINIA 23604

This report was prepared by The Boeing Company, Vertol Division, under the terms of Contract DAAJ01-71-C-0840(P6A). The objective of this effort was to define design criteria and to design, fabricate, and test simulated HLH aft rotor shaft tapered roller bearings to verify the capability of these bearings to sustain HLH rotor loads for 1200 hours.

The test results indicate that the actual HLH aft rotor shaft bearings' performance should be adequate and a life of more than 1300 hours should be achieved.

This directorate concurs with the conclusions presented herein.

The technical monitor for this effort was Mr. Wayne A. Hudgins, Heavy Lift Helicopter Project Office, Systems Support Division.

Contract DAAJ01-71-C-0840(P6A)
USAAMRDL Technical Report 73-92
January 1974

TEST RESULTS REPORT AND TECHNOLOGY DEVELOPMENT REPORT, HLH/ATC DRIVE SYSTEM AFT ROTOR SHAFT BEARING TEST

Final Report

By

J. W. Lenski, Jr

Prepared by

Boeing Vertol Company
A Division of The Boeing Company
Philadelphia, Pennsylvania

for

U.S. ARMY AVIATION SYSTEMS COMMAND
ST. LOUIS, MISSOURI

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FOREWORD

The tapered roller bearing development work presented in this report has been completed in partial fulfillment of Contract DAAJ01-C-0840(P40). This effort was to verify the capability of the HLH aft rotor shaft bearings to sustain HLH rotor loads for 1200 hours. The scope of the work included designing, manufacturing, and experimentally testing two sets of simulated HLH aft rotor shaft support bearings in a back-to-back test rig at loads equivalent to 112 percent of HLH aft rotor design loads.

This test program was conducted at the Boeing Vertol Company, under the technical direction of J. W. Lenski, Senior Design Engineer. Testing was conducted by H. E. Southerling and analytical support was provided by H. S. Faust. Technical direction was provided by Mr. W. Hudgins, Project Engineer, Eustis Directorate, U.S. Army Air Mobility Research and Development Laboratory, Fort Eustis, Virginia.

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LIST OF SYMBOLS

a	Shaft inner diameter, inch
b	Bearing bore diameter, inch
c	Outer diameter of bearing inner race small end, inch
d	Fatigue life ratio
I	Shaft fit, inch
K	Tapered roller bearing factor
L_{10}	Fatigue life which 90% of the bearing population endures, hours
P	Preload spacer length tolerance, inch
t	Operating time, hours

INTRODUCTION

In order to provide the most effective method of reacting the ever-increasing rotor loads of advanced heavy-lift helicopters, optimization of rotor shaft support bearing materials and design is required. Studies were conducted on various bearing configurations which are capable of supporting the thrust, radial, and moment loads produced in the helicopter rotor head. The studies showed that large-pitch-diameter and high-contact-angle, duplex-mounted tapered roller bearings resulted in the lightest weight and most compact bearing system for supporting the helicopter rotor shaft.

The HLH aft rotor shaft bearing application was designed for tapered roller bearings that achieved a minimum B-10 life of 2000 hours at 118,000 pounds gross weight and 6° flap angle. Three sets of bearings were designed and fabricated at the Timken Company, Canton, Ohio, to the specifications defined by Boeing Vertol.

After completion of the design and fabrication of the test bearings several design changes were made concerning the HLH transmission bearings which resulted in dimensional changes to the rotor shaft bearings. A comparison of the test bearings and the actual transmission bearing is shown on Table I. These changes basically were the results of weight savings effort and design load changes. The test bearings used in this program simulated all the major design factors and therefore were tested to evaluate the HLH rotor shaft bearing application. The result of this test program can be used to approximate the performance and expected fatigue life of the actual HLH rotor shaft bearings.

The test bearings were operated in a back-to-back test rig in a simulated rotor shaft bearing housing and under simulated overload condition. Testing was scheduled for 1200 hours or to the failure of two bearings. Various operating parameters were evaluated during this test program, and the results obtained were directly applicable to the actual HLH rotor shaft bearings. The results of this test program will be summarized in this report.

TABLE I. COMPARISON OF TEST BEARING AND ACTUAL HLH TRANSMISSION BEARINGS

	Upper Bearing		Lower Bearing	
Bearing Location	Test Rig	HLH Transmission	Test Rig	HLH Transmission
Boeing Vertol P/N	SK301-10257-2	301-10406-1	SK301-10257-1	301-10409-1
Contact Angle, deg	35	35	45	45
Pitch Diameter, in.	23.5	23.4	24.0	24.08
Number of Rollers	61	66	52	56
Roller Diameter, in.	1.162	1.052	1.435	1.325
Roller Length, in.	1.460	1.120	1.829	1.423
Radial Capacity, lb	44,000	31,400	50,000	36,400
Thrust Capacity, lb	78,500	56,000	128,000	93,500
Cage Design	Pin Type	Pin Type	Pin Type	Pin Type
Roller Spherical End Radius, pct	80	80	80	80
Weight, lb	87.7	60	137.5	98

TECHNICAL APPROACH

BACKGROUND

The HLH aft rotor shaft support bearings B-10 life requirements are directly influenced by the rotor head loads and the type of bearing configuration used to react these loads. To minimize the size and weight of these rotor shaft support bearings, the highest capacity bearing configuration must be used. Trade studies were conducted to determine the most efficient method of reacting the HLH rotor loads. The design study was conducted using the following ground rules:

1. Bearings to react rotor loads based on 118,000 pound gross weight and 6° flap angle

2. Rotor head loads for the above condition are:

Thrust	70,800	pounds
Drag	15,030	pounds
Moment	872,000	inch-pounds
Speed	156	rpm

3. Minimum bearing B-10 life of 2000 hours under these conditions

4. Bearing pitch diameter to be approximately 24 inches

5. Select optimum bearing configuration for minimum weight and size and maximum life

The bearing configuration which meets the above criteria was a duplex-mounted tapered roller bearing (Figure 1) which had pitch diameters of 23.5 and 24.0 inches and contact angles of 35 and 45 degrees. In addition, these bearings were designed with pin-type cages which provided for maximum capacity within a given cross-sectional area. The pin-type cage allows for the rollers to be guided by pins through a hole in the center of each roller and allows for minimum space between rollers so that the maximum roller complement can be achieved.

The use of tapered roller bearings to support combined thrust, radial, and moment loads is not new; but because of several departures from conventional bearing size and geometry configuration, a test program was conducted to evaluate the effect of the following critical parameters:

1. Effects of high contact angle on bearing performance.

2. Effects of azimuth variations in stiffness of the bearing housing on the internal load distribution of the bearings.

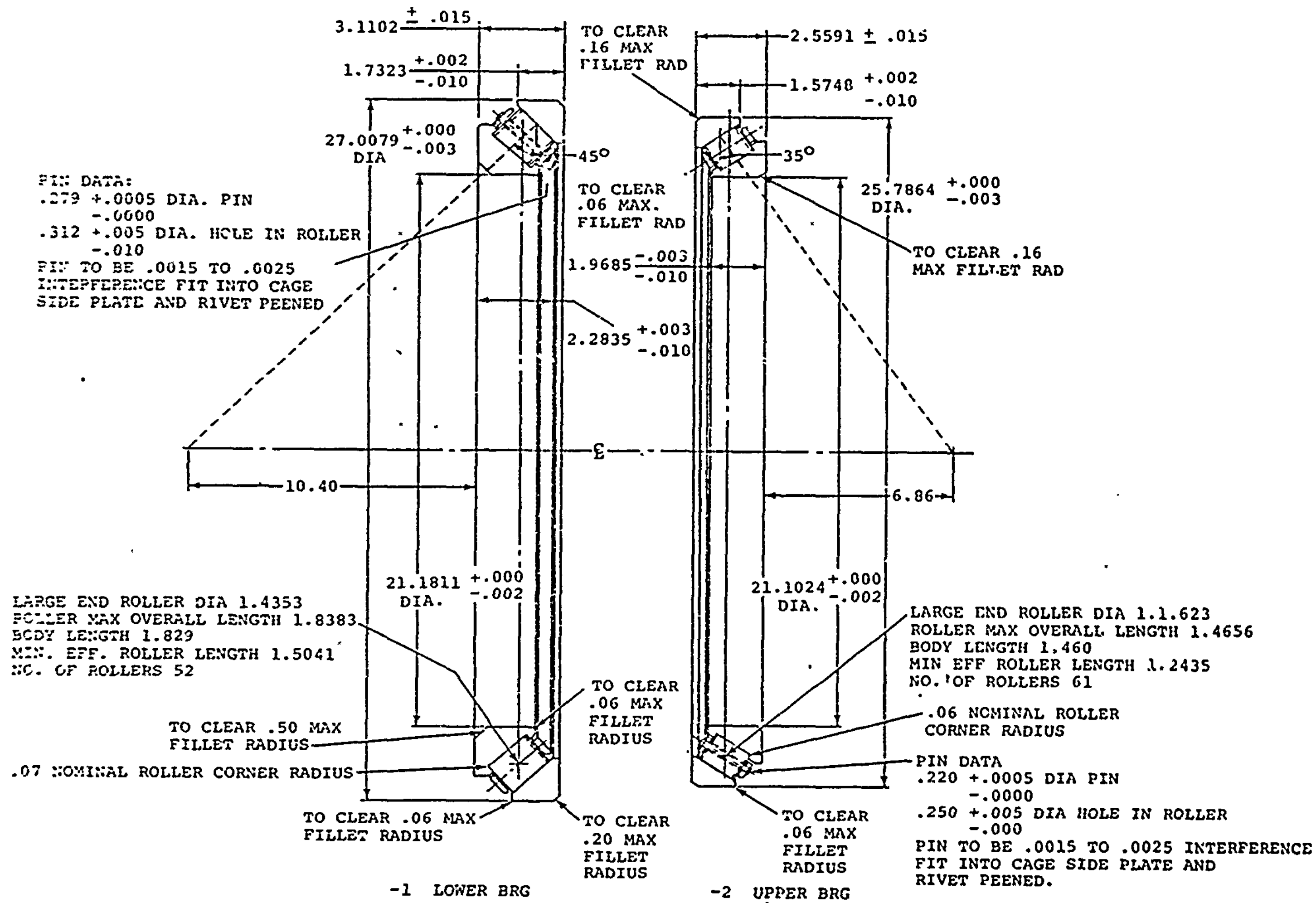

Figure 1. Details of Test Specimen Design and Description.

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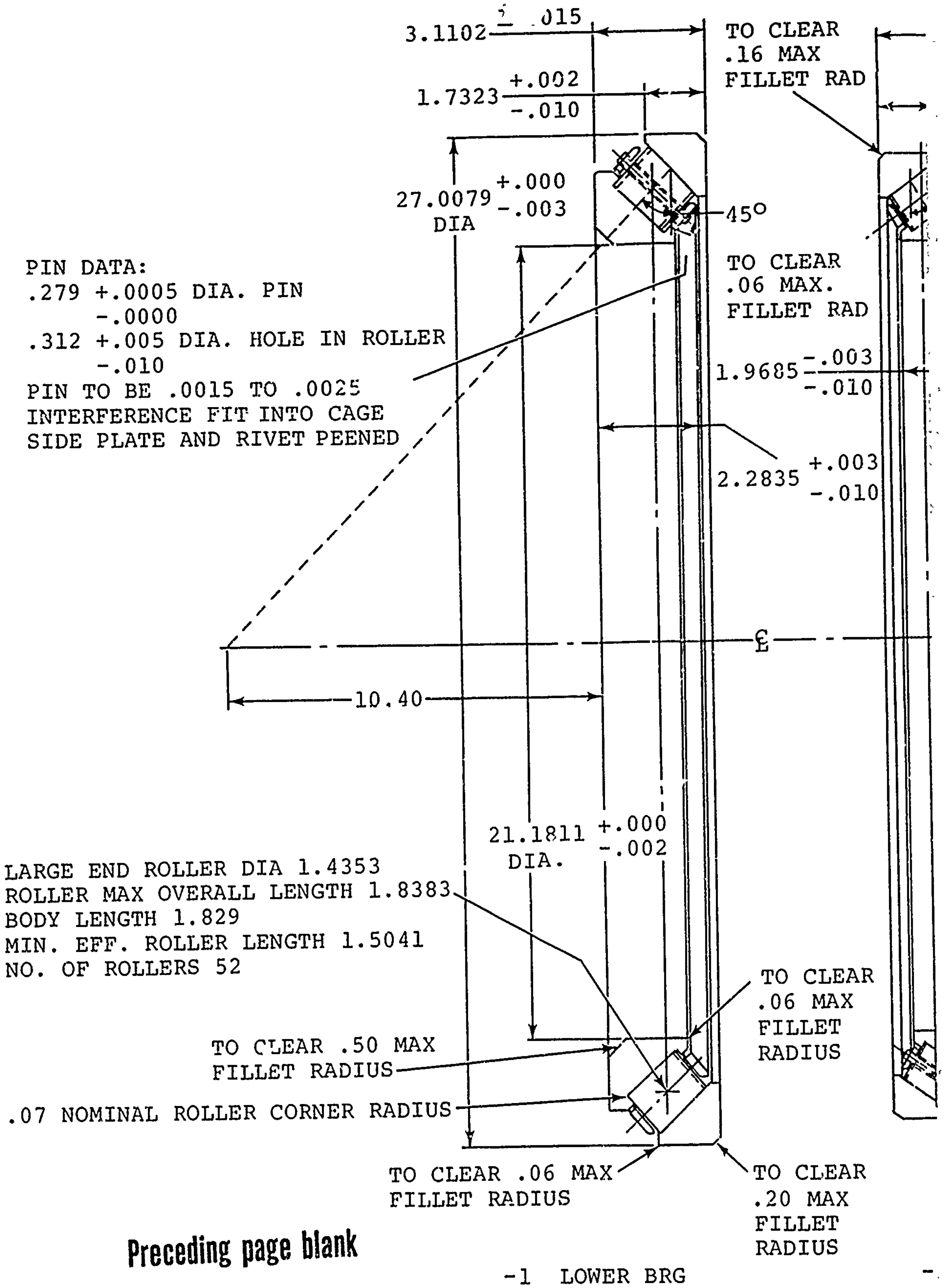

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Figure 1. Details of Test Specimen Design and Description.

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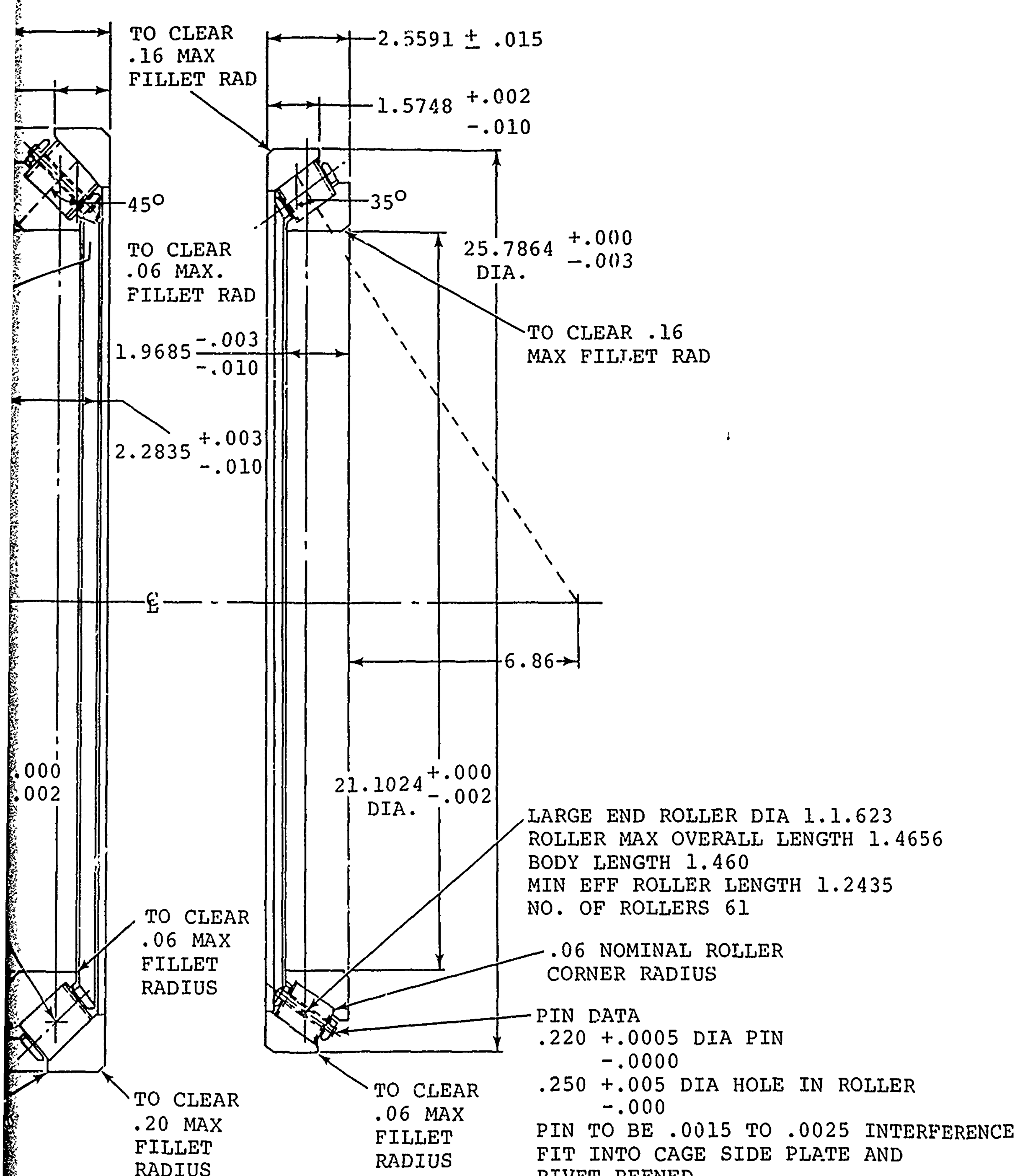

R BRG -2 UPPER BRG

and Description.

B

3. Effects of various preloads on bearing performance.

4. Effects of low oil film thickness on bearing life.

5. Effects of pin-type cage on bearing performance and life.

As the contact angle of a tapered roller bearing increases, the reaction load of the roller end on the cone rib flange also increases. The HLH rotor shaft bearing test program will determine if this increased flange load will have any effect on the wear or scoring hazard of the flange surface. Review of the past experience and test results of shallower angle tapered roller bearings indicate that the conditions present on the HLH rotor shaft bearing should not be critical.

One of the most important factors controlling fatigue life, and the most difficult to analyze, is the effect of variation of housing stiffness on the internal load distribution of the bearing. The bearing housing is supported to the aircraft structure by four legs. This type of housing configuration results in four areas of high stiffness. Present bearing life analyses are based on the assumption that the bearing housing is uniformly rigid. The effect of variable stiffness will result in locally high peak loads which may reduce fatigue life.

A proper preload for the rotor shaft bearing will be established by test. Higher contact angle bearings are more sensitive to preload, and excessive preload may produce excessive heat generation and reduced bearing life. Several preload conditions will be evaluated to determine which provides the required system stiffness and still maintains an acceptable heat generation and adequate life.

Because of the heavy loading expected on these bearings and the low viscosity of the lubricant, lubricant film thickness may affect bearing fatigue life by causing surface distress. Lubricant film thickness calculations were made on a comparative basis for the CH-46 forward rotor shaft Timken bearings (shallow angle) and the proposed HLH rotor shaft bearings (SK301-10257). The results are summarized on Table II.

From Table II, it can be concluded that the HLH bearings, from the viewpoint of film thickness, are better than the CH-46 bearings. Even though the film thickness is increased, it is very small compared to surface roughness and may affect bearing life.

The use of a pin-type cage design allows for an increase of approximately 15 percent in the number of rollers in each bearing over that of a standard pressed steel cage used in tapered roller bearings. The larger number of rollers increase the load-carrying capability by 10 percent and the fatigue life by

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approximately 38 percent. This type of cage design has not been used in this type of application, and therefore the design will be evaluated for pin wear and strength.

TABLE II. COMPARISON OF HLH VERSUS CH-46 ROTOR SHAFT BEARING OPERATING CONDITION

	CH-46D A02DS273-6 (Upper)	CH-46D A02DS273-5 (Lower)	HLH Upper	HLH Lower
Radial Load, lb	13,615	11,550	63,280	48,240
Thrust Load, lb	0	11,820	0	70,800
Speed, rpm	264	264	156	156
Lubricant	MIL-L-23699 →			
Temperature, °F	190 →			
Film Thickness, microinches	1.55	2.3	3.39	3.45

The above parameters were evaluated in a back-to-back test rig which was designed and fabricated by Boeing Vertol to simulate the expected loading, lubrication, housing stiffness, and preload of the HLH aft rotor shaft bearing application. Approximately 1100 hours of endurance testing was conducted to evaluate bearing performance and adequacy of bearing life under predicted operating conditions.

STATEMENT OF PROBLEMS

Presently there are several conventional bearing configurations (ball thrust and radial roller bearing, and duplex tapered roller bearing) which can be used to support the rotor hub loads of a helicopter. With ever-increasing rotor hub loads, the use of these conventional bearing systems does not provide for the most optimum bearing arrangement from the standpoint of life, weight, and size.

Trade studies have shown that large-pitch-diameter, steep-contact-angle tapered roller bearings could provide the highest fatigue life for minimum weight and size. The only objection to the bearing configuration is the lack of test experience of large-diameter, steep-angle tapered roller bearings to support helicopter rotor loads. Therefore, the primary concern with this program is to determine the feasibility of using this

type of bearing configuration. In addition, several other factors are to be evaluated concerning bearing performance:

1. Effect of azimuth housing stiffness variation on bearing fatigue life
2. Effect of preload
3. Effect of low oil film thickness on bearing fatigue life
4. Pin-type cage design

Two sets of bearings were tested in a back-to-back test rig which was designed to simulate the expected loading, lubrication system, housing stiffness, and preload of the HLH aft rotor shaft bearing application. Test loads were established to approximate 112 percent loading. The test rig loading condition was as follows:

Thrust	70,800 lb
Moment	2,175,000 in.-lb
Speed	156 rpm
Lubricant	MIL-L-23699
Oil Inlet Temperature	190°F

The results of this test will determine the feasibility of using large-pitch-diameter, steep-contact-angle tapered roller bearings to react and support the expected HLH rotor hub loads.

TEST METHOD

DESCRIPTION OF TEST SPECIMENS

Six test bearing specimens were manufactured by the Timken Company of Canton, Ohio. The test bearings used in this program conformed to the physical envelope and capacity of the bearings defined on Boeing Vertol drawings SK301-10257-1 and -2. Complete dimensional details and fabrication requirements are shown in Figure 1.

Bearing pitch diameters were 23.5 inches and 24 inches for the upper and lower bearings, respectively, with corresponding contact angles of 35 degrees and 45 degrees. Use of the pin-type cage allows for an increase of approximately 15 percent in the number of rollers in each bearing over that of a standard pressed-steel cage, thereby increasing the load-carrying capability and fatigue life. The upper bearing had 61 rollers and the lower bearing had 52 rollers. The cones and cups of each bearing were fabricated from consumable electrode vacuum melted SAE 3310 steel also with a surface hardness of Rc 58 to 63.

The specimens submitted for testing are identified on Table III. The bearings were tested with the cup and cone combinations shown.

TABLE III. IDENTIFICATION OF HLH ROTOR SHAFT TEST BEARINGS

Bearing	Boeing Vertol P/N	Timken P/N Cup	Timken P/N Cone	Timken S/N Cup	Timken S/N Cone
Upper	SK301-10257-2	XC10837D	XC10837C	72-4	72-4
				72-3	72-3
				72-1	72-1
Lower	SK301-10257-1	XC10838D	XC10838C	72-8	72-4
				72-6	72-2
				72-5	72-1

TEST APPARATUS

Testing was conducted in a test fixture (Figure 2) designed to test two sets of bearings simultaneously in a back-to-back configuration in simulated aircraft bearing housing assemblies. The test machine was designed to test the aft rotor shaft bearings at a rotor speed of 156 rpm, while subjected to a thrust load of 70,800 pounds and a moment of 2,175,000 inch-pounds. The test specimen housings were designed to simulate the transmission upper cover. Lubrication was supplied by jets. Four jets provided lubricant to the small end of each bearing and two jets provided lubrication to the larger end of the upper bearing of each set. The prominent features of the test fixture are shown in Figures 3 and 4 .

In order to assist in removing the bearings from their seats on the shaft, each bearing seat is provided with an oil groove which is pressurized from inside the drive shaft (Figure 5). By pressurizing the oil groove, a radial force is generated which relieves some of the load due to the interference fit between shaft and bearing. While the oil pressure is applied to the annulus, the bearings can be removed from the shaft by lifting in a vertical direction as shown in Figure 5 . In addition to providing a medium for relieving the interference fit between the bearings and the shaft, the hydraulic fluid also acts as a lubricant at the bearing/shaft interface to protect the components during disassembly.

Loading was accomplished hydraulically by a means of a "load maintainer" which has the ability to automatically maintain the desired load. An oil heater (circulation type) was used to preheat the lubricant prior to entering the test bearing cavities.

TEST PARAMETER INSTRUMENTATION

Test loads were measured at points shown in Figure 4 using four-arm, strain-gaged tension/compression links and 1/4° accuracy pressure gages. Oil inlet temperature was monitored by means of a thermocouple located between the outlet of the oil heater and the inlet to the test bearing cavities. The oil outlet temperatures were monitored in the oil return lines for the upper and lower bearing assemblies. All temperature data was recorded on a multichannel temperature recorder. Each of the bearing housings was provided with two accelerometers to detect peak "g" levels. Chip detectors were installed in the oil return lines of each bearing assembly housing. Total accumulated test running time was indicated with an electric timer. Figures 2 and 6 show the type of instrumentation and the location of the sensors. Total test specimen lubricating oil flow was measured using a turbine-type flowmeter, located in the lubricating oil supply line, in conjunction with a

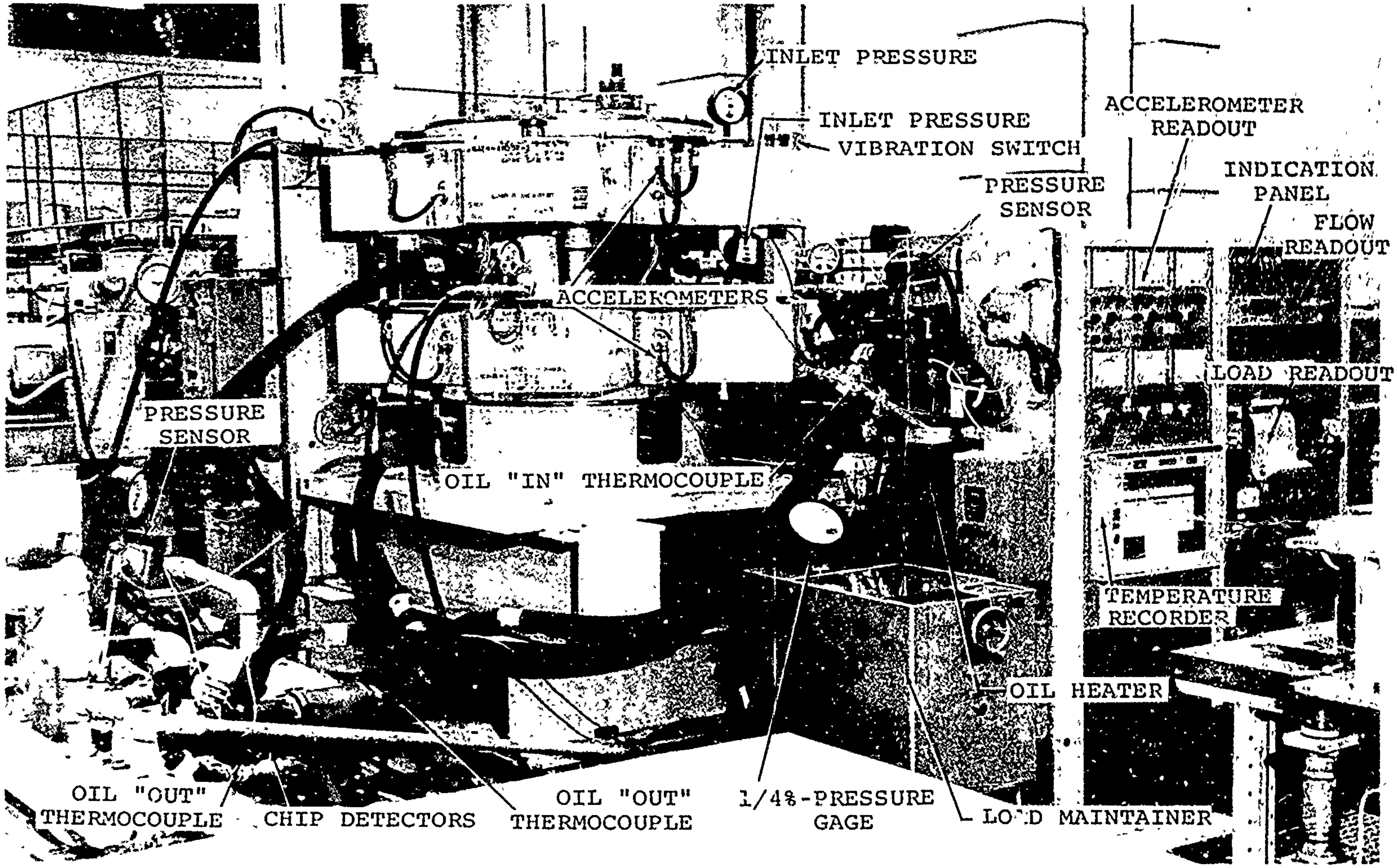


Figure 2. HLH Aft Rotor Shaft Bearing Test Rig.

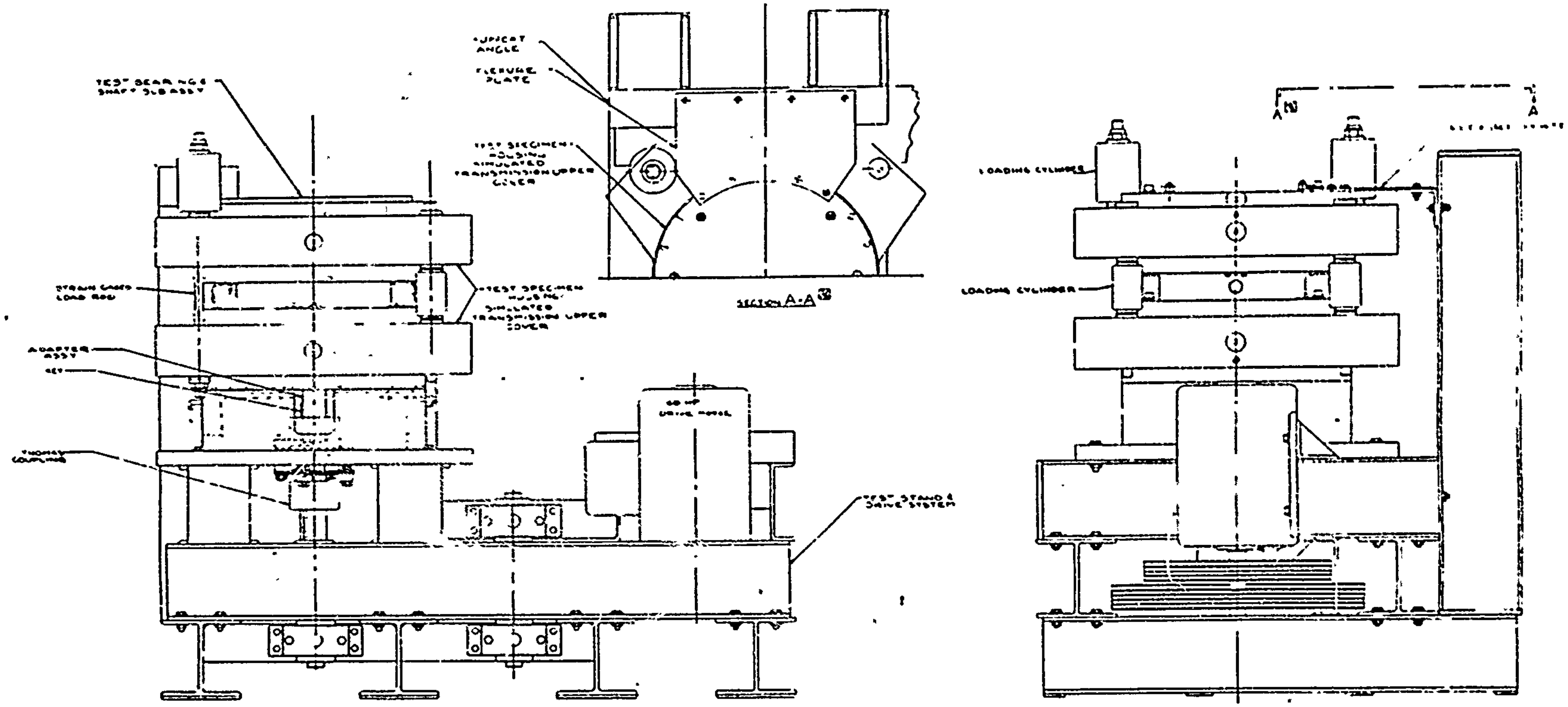


Figure 3. Test Stand Assembly.

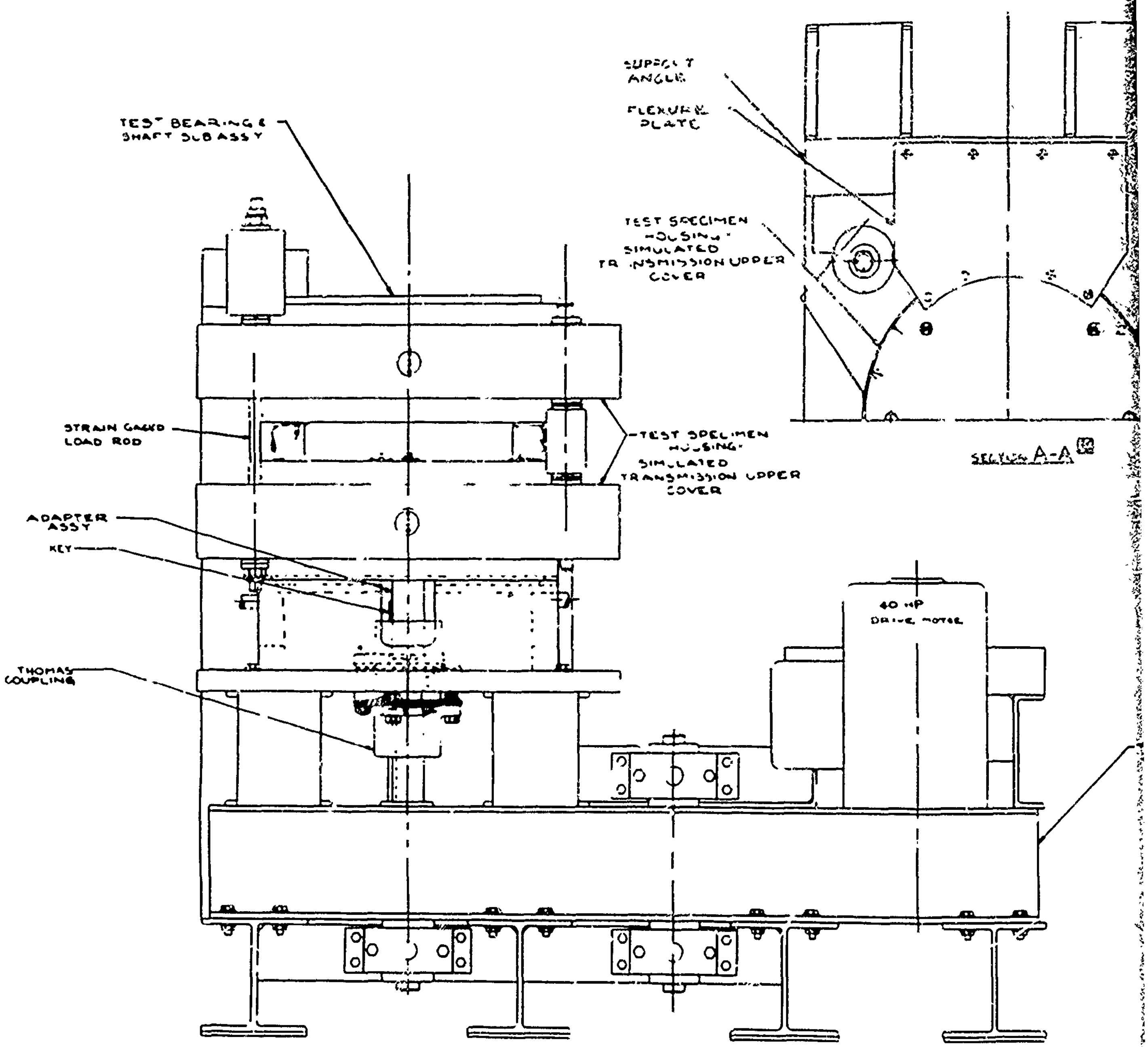
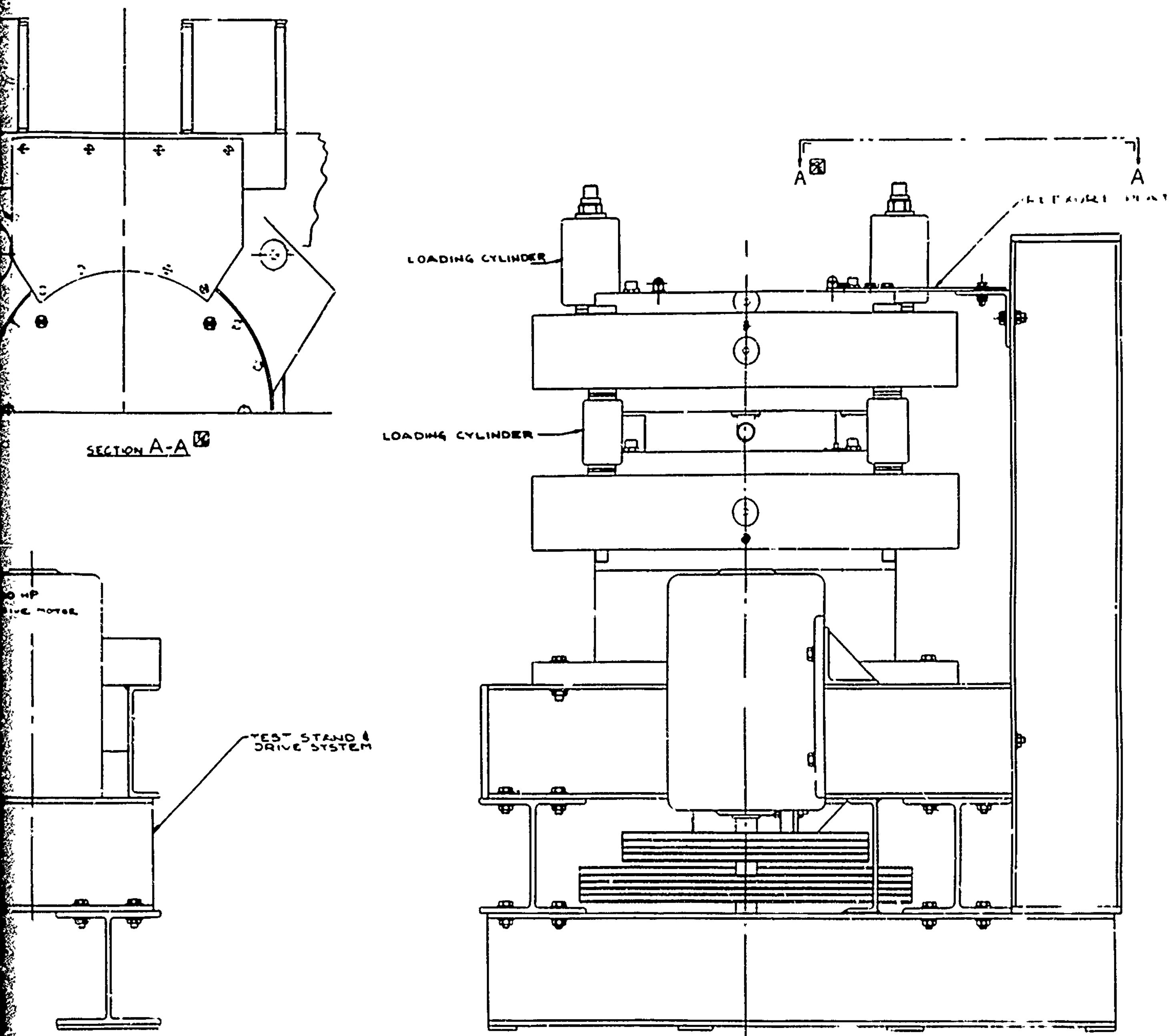


Figure 3. Test Stand Assembly.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LOADING CYLINDER
LOADING CYLINDER
SECTION A-A
TEST STAND & DRIVE SYSTEM

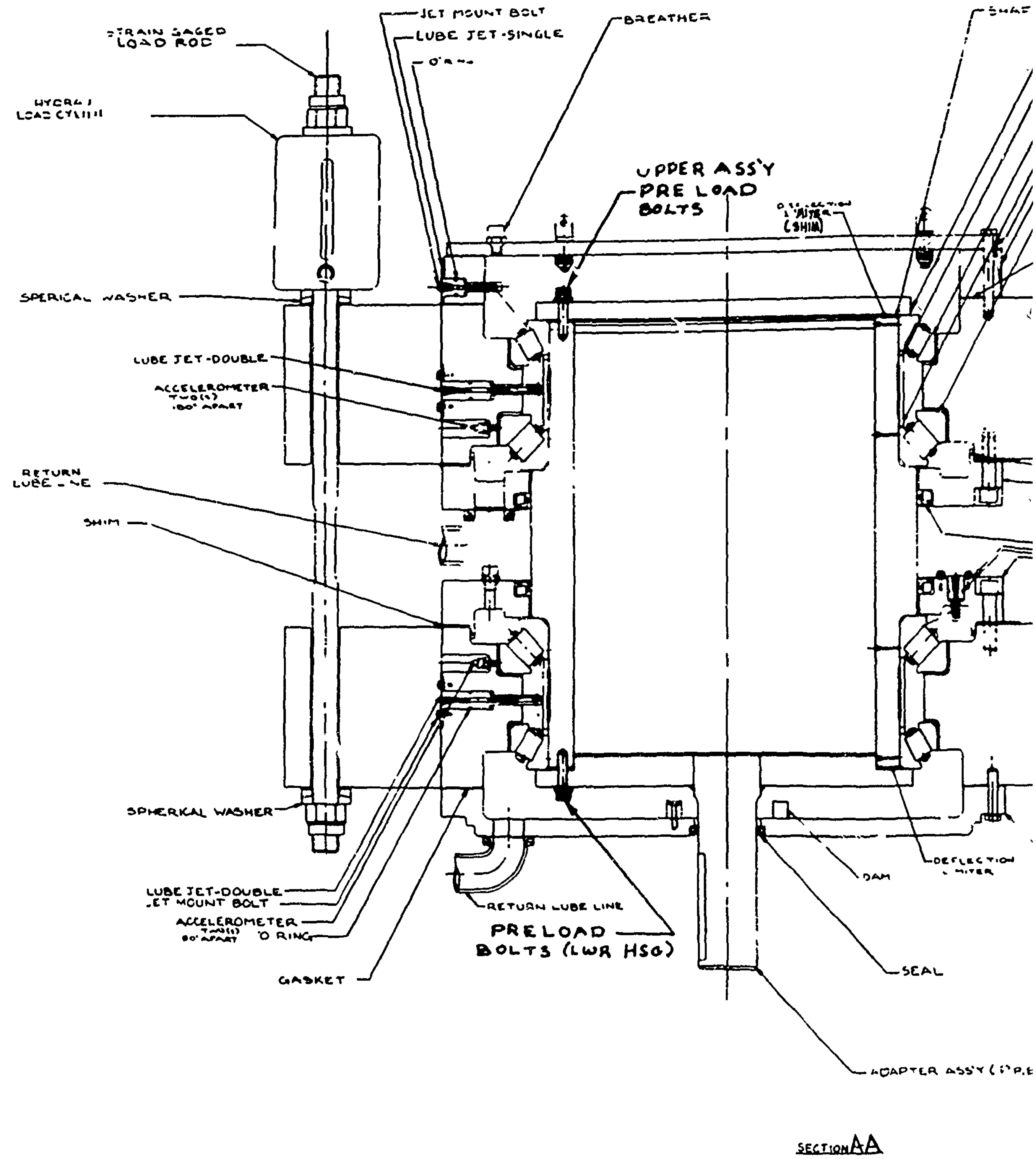

Figure 4. Shaft and Bearing Assembly.

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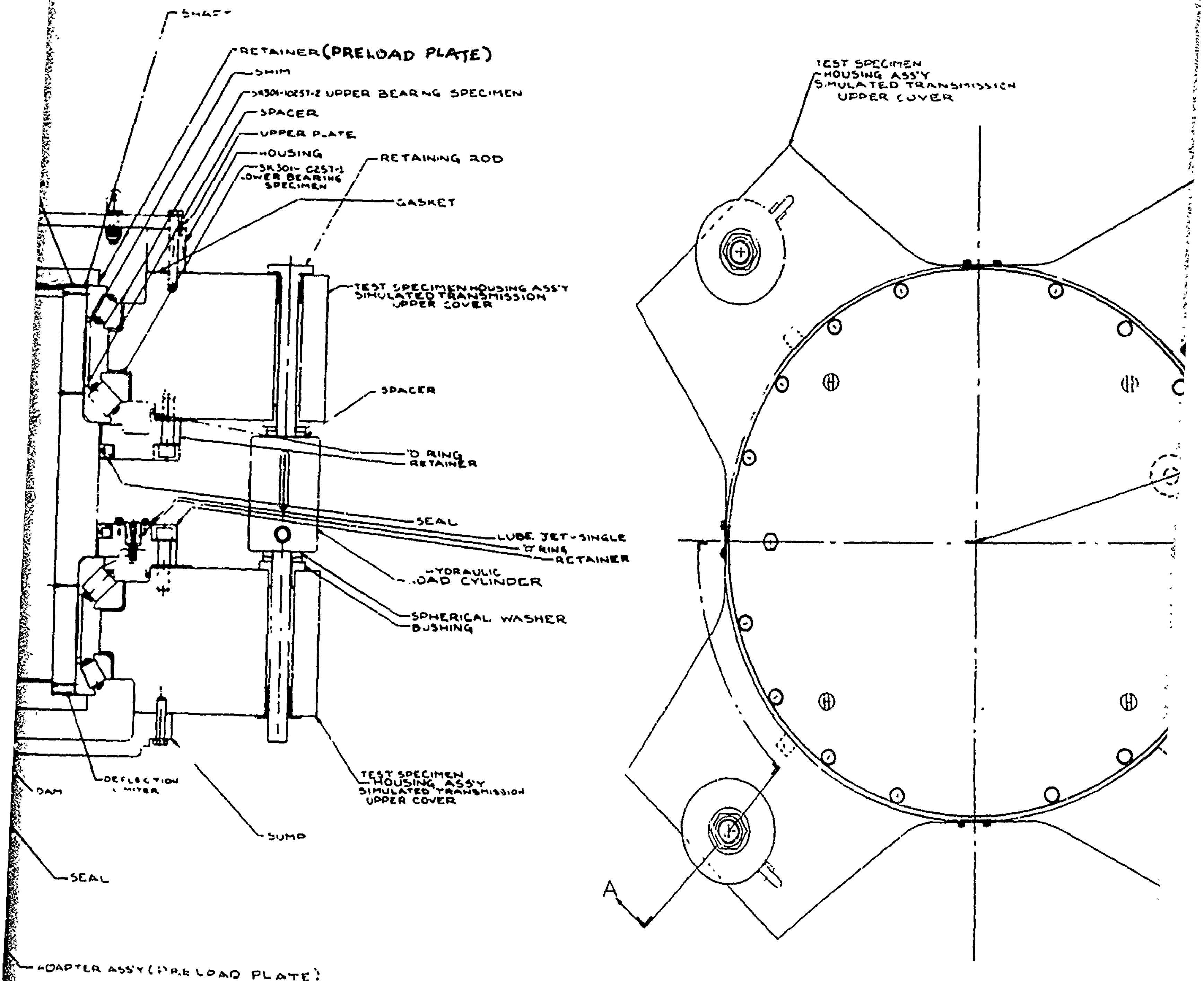
SHAFT
RETAINER (PRELOAD PLATE)
SHIM
SK301-10257-2 UPPER BEARING SPECIMEN
SPACER
UPPER PLATE
HOUSING
SK301- 0257-1 LOWER BEARING SPECIMEN
RETAINING ROD
GASKET
TEST SPECIMEN HOUSING ASS'Y SIMULATED TRANSMISSION UPPER COVER
SPACER
O RING
RETAINER
SEAL
LUBE JET-SINGLE
O RING
RETAINER
HYDRAULIC LOAD CYLINDER
SPHERICAL WASHER
BUSHING
TEST SPECIMEN HOUSING ASS'Y SIMULATED TRANSMISSION UPPER COVER
DEFLECTION LIMITER
DAM
SUMP
SEAL
ADAPTER ASS'Y (PRELOAD PLATE)
TEST SPECIMEN HOUSING ASS'Y SIMULATED TRANSMISSION UPPER COVER
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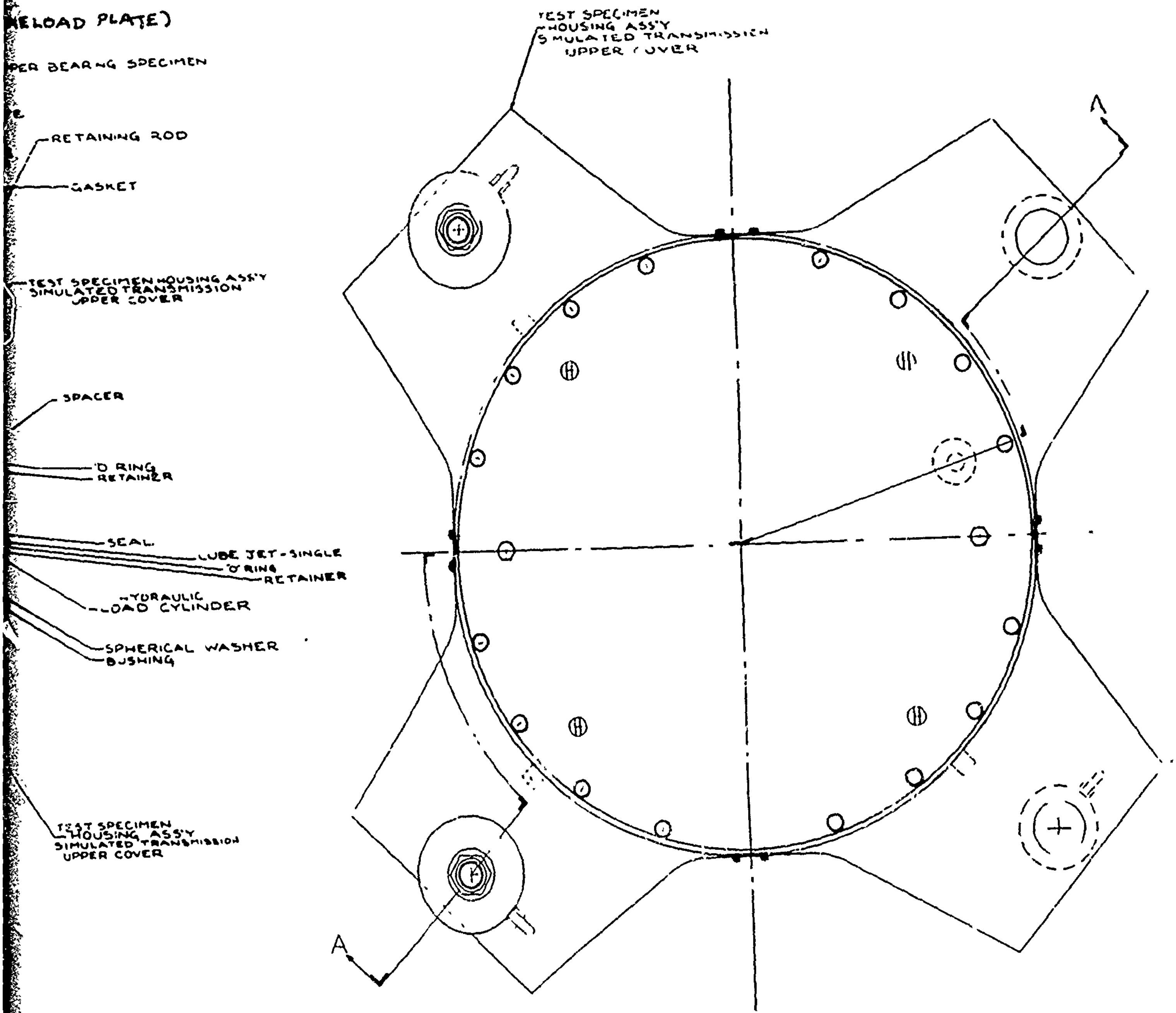

(PRELOAD PLATE)
PER BEARING SPECIMEN
RETAINING ROD
GASKET
TEST SPECIMEN HOUSING ASS'Y
SIMULATED TRANSMISSION
UPPER COVER
SPACER
'O' RING
RETAINER
SEAL
LUBE JET - SINGLE
'O' RING
RETAINER
HYDRAULIC
LOAD CYLINDER
SPHERICAL WASHER
BUSHING
TEST SPECIMEN
HOUSING ASS'Y
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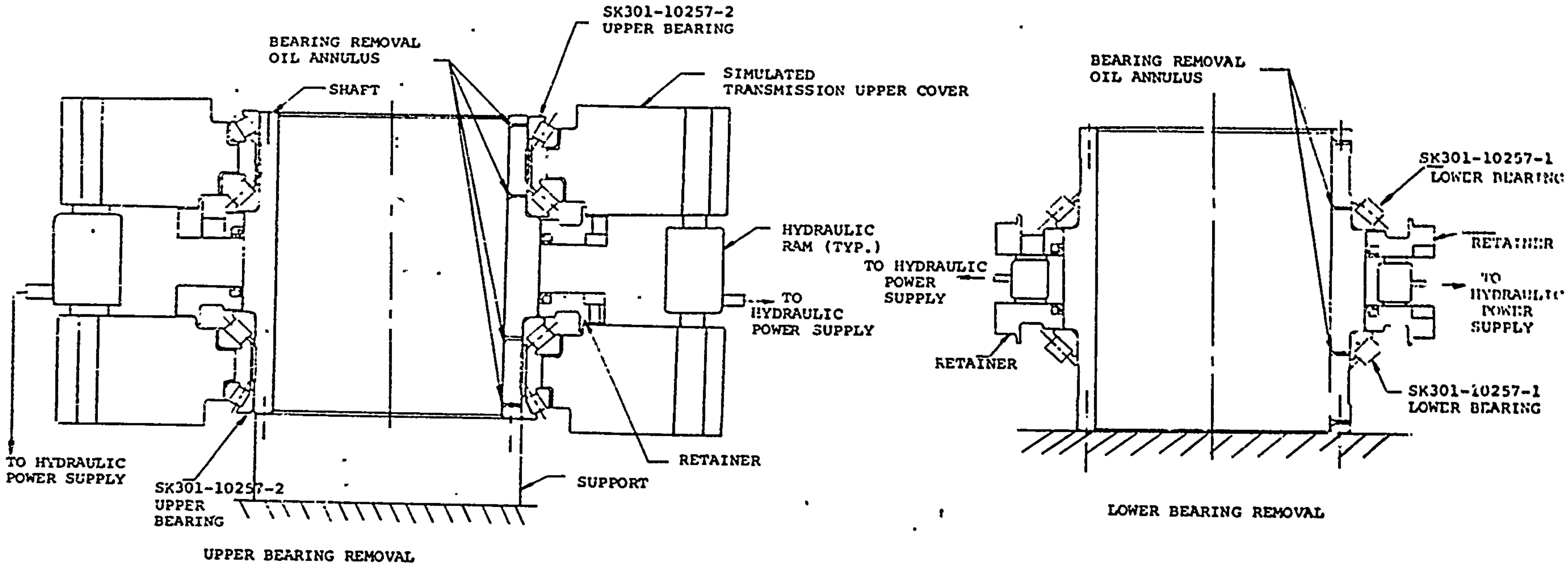

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Figure 5. Test Bearing Removal Setup.

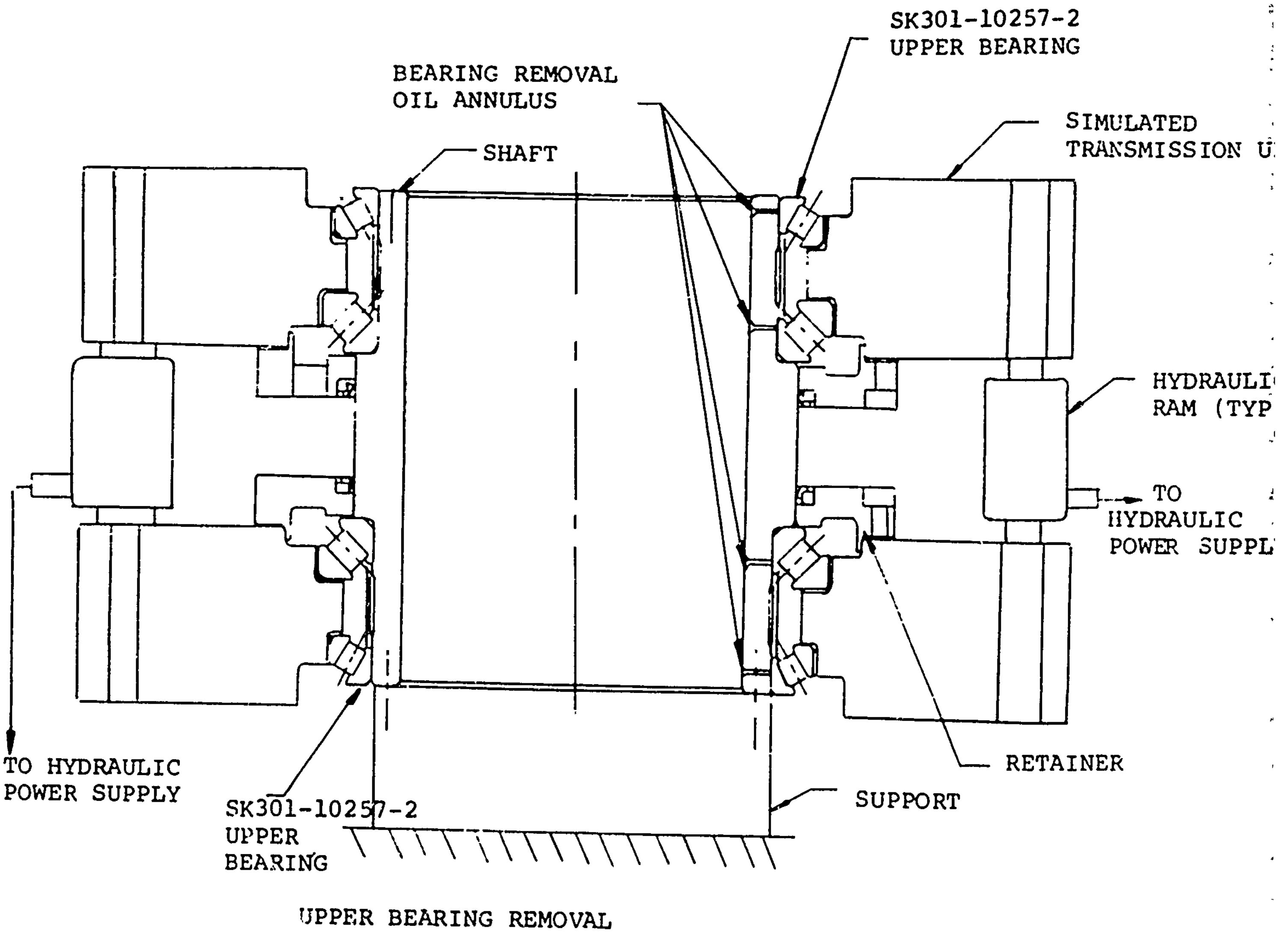

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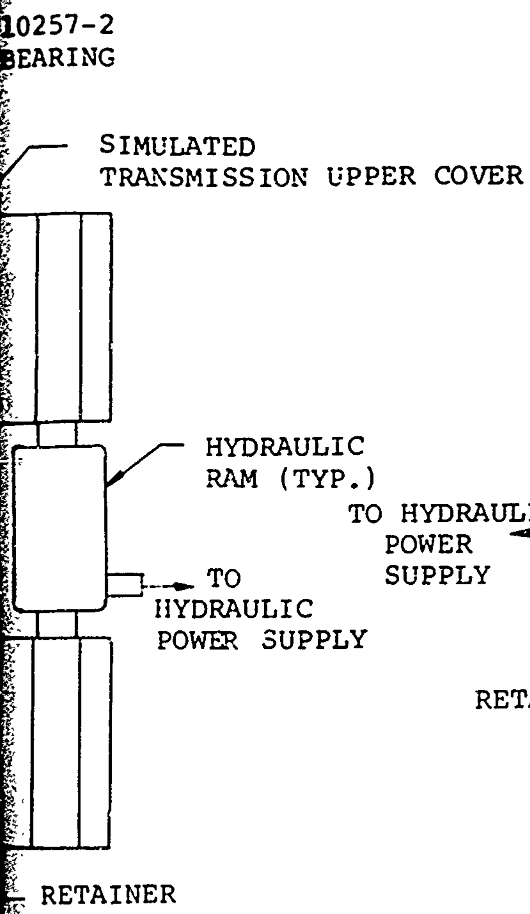
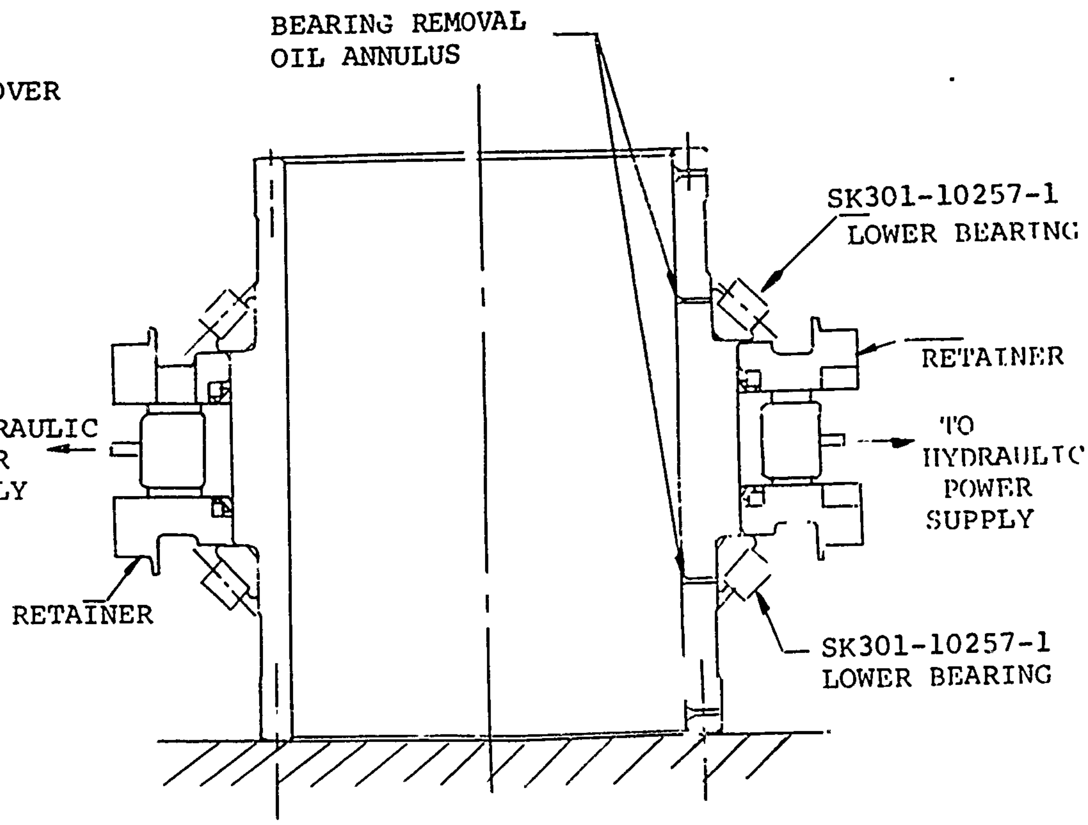
Figure 5. Test Bearing Removal Setup.





LOWER BEARING REMOVAL

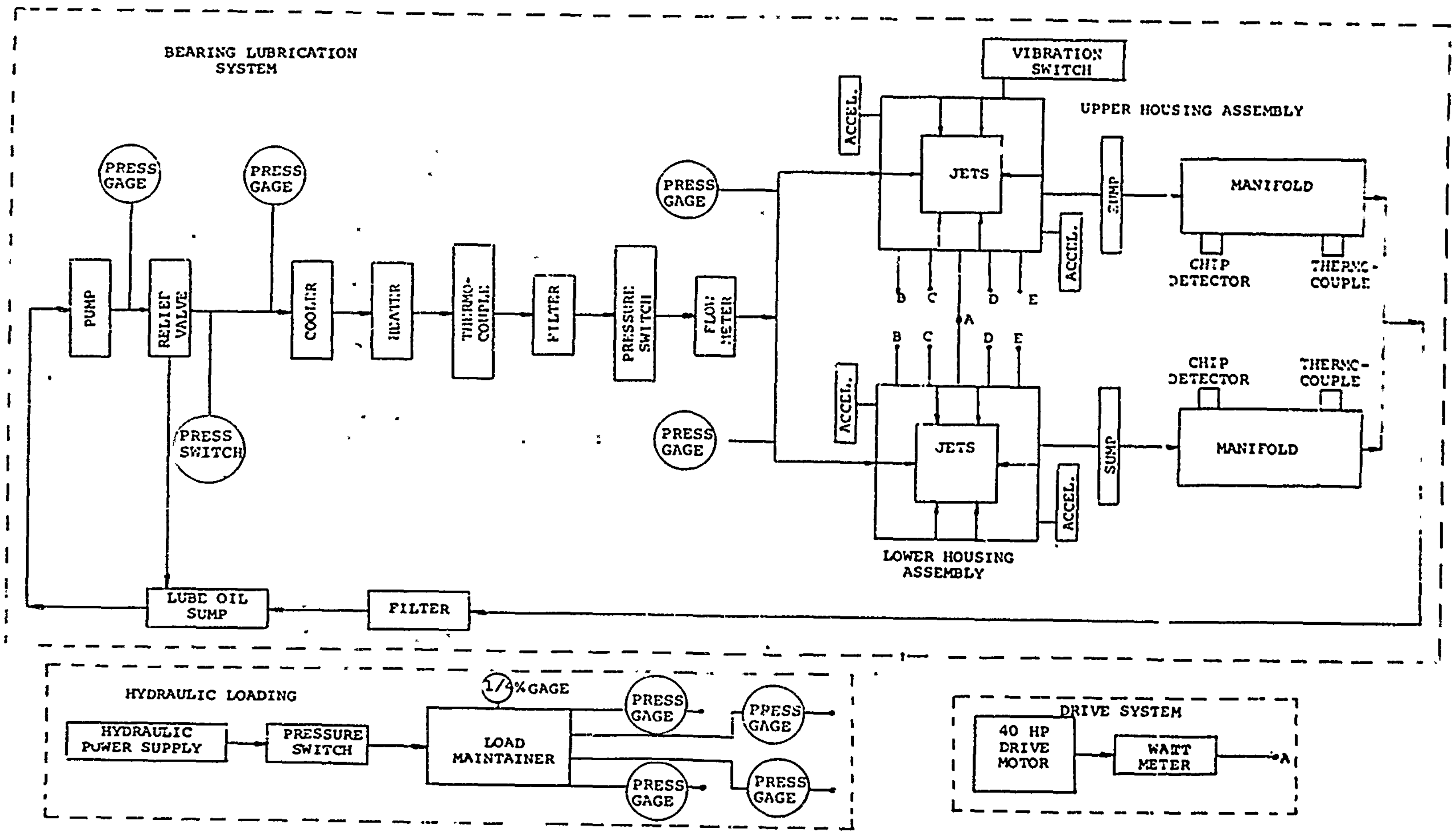


Figure 6. Test System.

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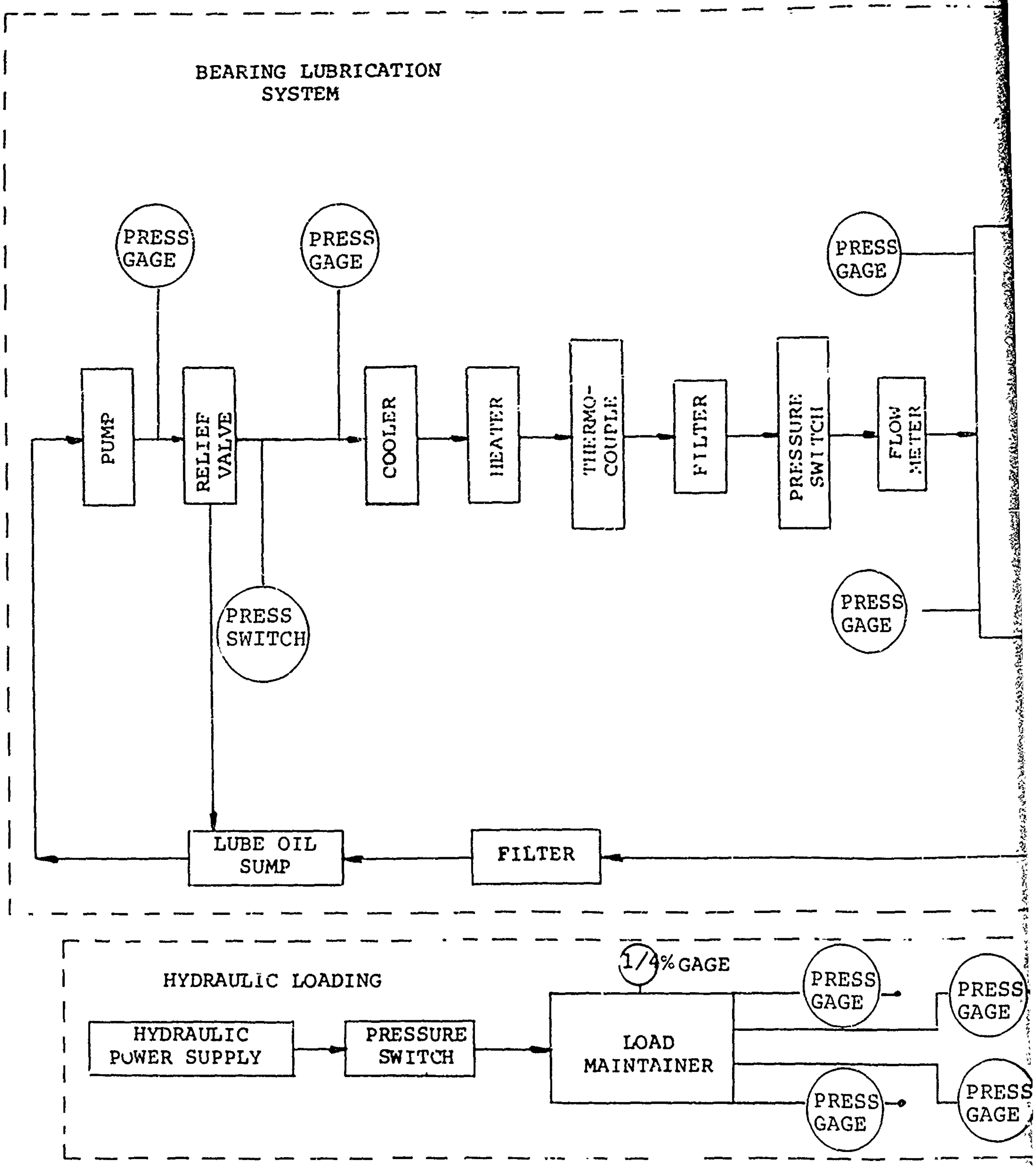

Figure 6. Test System.

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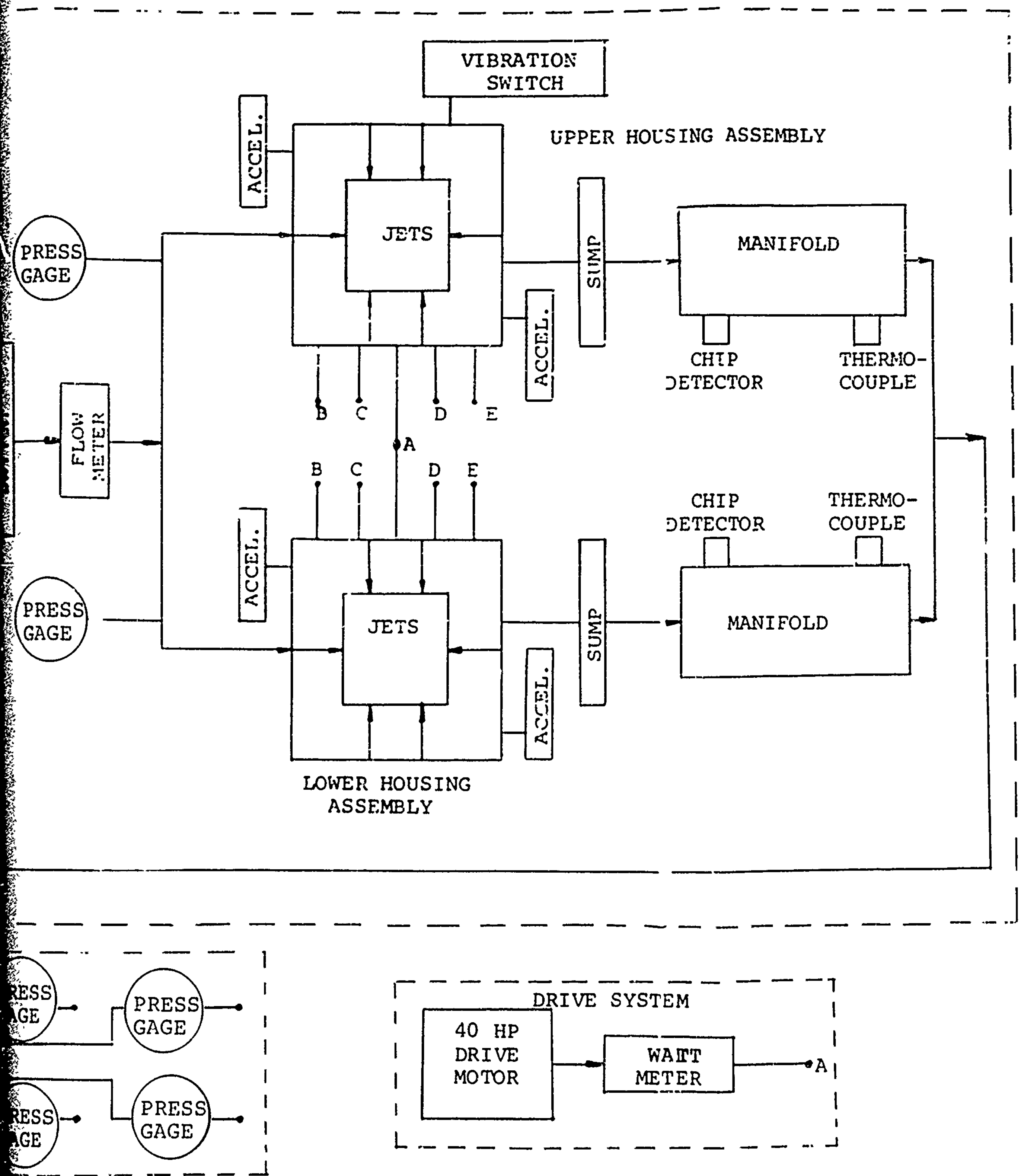

VIBRATION SWITCH
UPPER HOUSING ASSEMBLY
ACCEL.
JETS
PRESS GAGE
SUMP
MANIFOLD
ACCEL.
CHIP DETECTOR
THERMO-COUPLE
FLOW METER
B C D E
A
B C D E
CHIP DETECTOR
THERMO-COUPLE
ACCEL.
PRESS GAGE
JETS
SUMP
MANIFOLD
ACCEL.
LOWER HOUSING ASSEMBLY
PRESS GAGE
PRESS GAGE
DRIVE SYSTEM
40 HP DRIVE MOTOR
WATT METER
A
PRESS GAGE
PRESS GAGE

B

frequency counter. Total test system power consumption was measured with a watt meter installed at the drive motor output.

TEST FAIL-SAFE INSTRUMENTATION

In order to protect the test system from catastrophic failure, various fail-safe devices were employed to shut down the test if test parameters fell outside preset levels. Fixture vibration was sensed with a "vibraswitch" mounted on the upper cover. Excessive fixture vibration resulted in the "vibraswitch" shutting down the test.

Pressure switches were employed to detect loss of hydraulic pressure used to apply the test load, loss of test bearing lubricating oil at the source, and loss of lubricating oil pressure at the test bearing input points. Loss of pressure at any of these locations resulted in test machine shutdown.

In addition to recording various temperatures, the multichannel temperature recorder provided a shutdown function in the event the inlet oil temperature was too high (indicative of oil heater malfunction) or the outlet oil temperature was too high (indicative of test bearing failure).

Accumulation of metallic chips on the chip detectors would also result in test machine shutdown. In the event of a machine shutdown, the cause of the shutdown is indicated on a lighted panel.

The allowable deviations of the test parameters are given in Table IV. Figures 2 and 6 show the locations of the various fail-safe devices.

ENDURANCE TEST PROCEDURE

Testing was conducted in a test fixture (Figure 2) designed to test the bearings in a back-to-back configuration in simulated aircraft bearing housing assemblies. The test bearing cones (inner races) were installed on the shaft by heating the bearings to 275°F. The cups (outer races) were soaked in carbon dioxide (CO_2) prior to installation into their housings. The test shaft was manufactured from 9310 steel per AMS 6250 (Rc 34-38). Testing was scheduled for 1200 hours under the conditions shown in Table IV which is equivalent to 112 percent aircraft design loads. The application of the required test loads is shown in Figure 7. Analysis of the establishment of the test loads is shown in Figure 8. The testing actuator loads were set at approximately the test rig limitation and provided the lowest B-10 life that could be achieved on this test rig. If these bearings were used in the HLH aircraft, each hour of testing would be equivalent to approximately 3 hours of aircraft time at 6° flap.

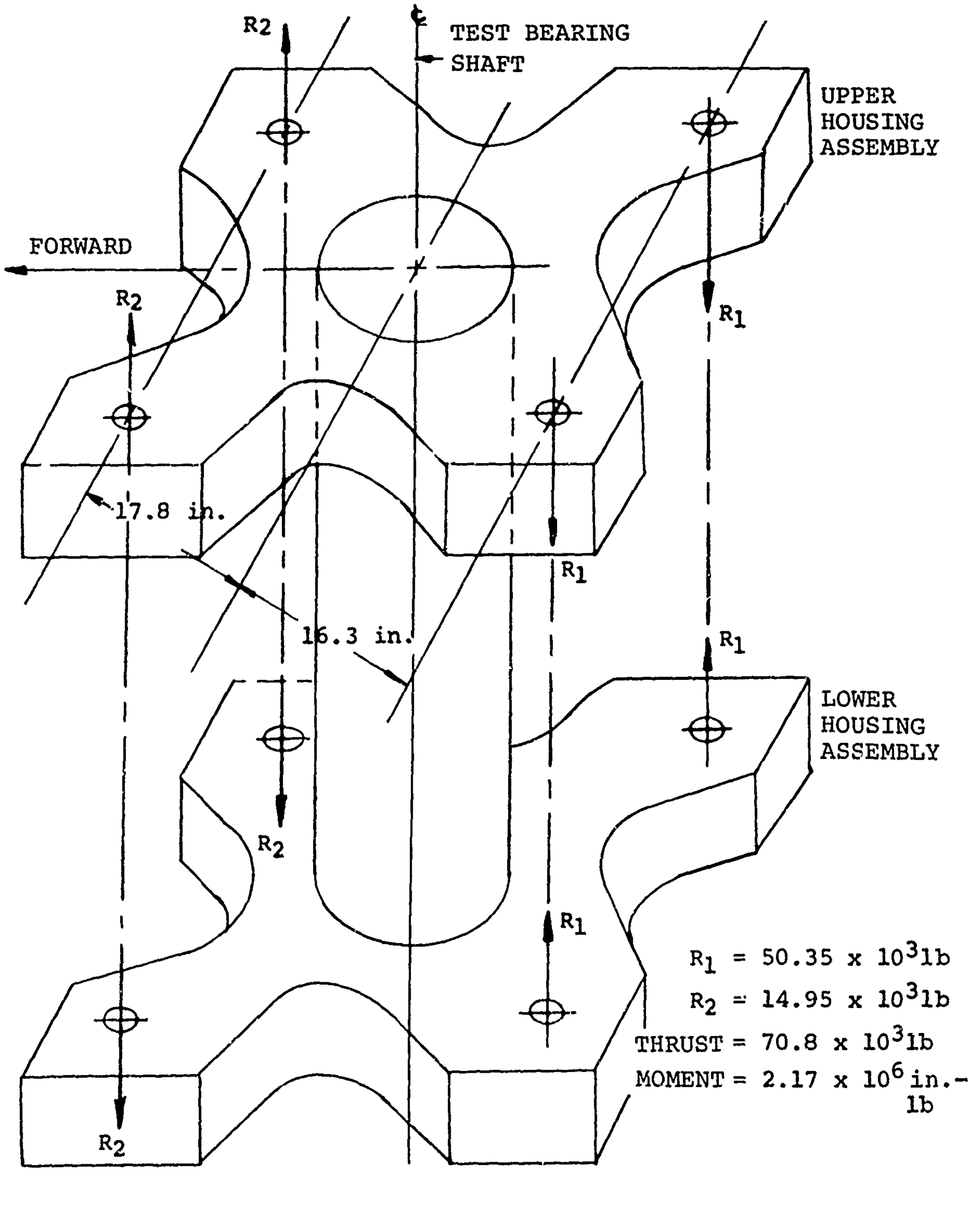


Figure 7. Aft Rotor Shaft Test Loading Schematic.

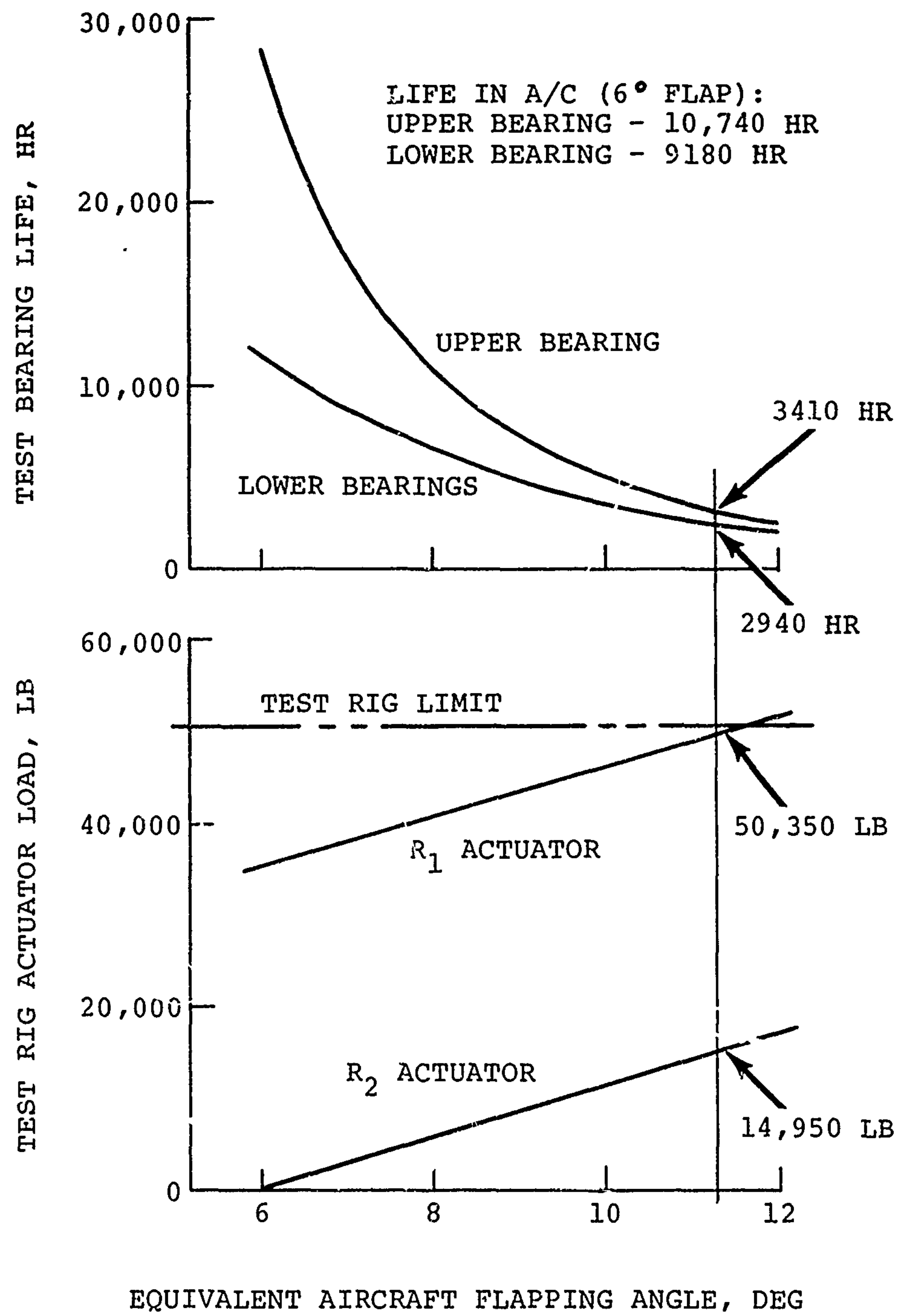


Figure 8. Selection of Test Loads for
Rotor Shaft Bearing Test.

TABLE IV. HLH ROTOR SHAFT BEARING TEST CONDITIONS	
Loads	70,800 lb thrust 2,175,000 in.-lb moment
Speed	156 rpm
Lubricant	MIL-L-23699
Oil Inlet Temperature	190°F $\pm$10°F
Oil Inlet Pressure	50 psig $\pm$5 psig
Oil Flow	3 gpm total

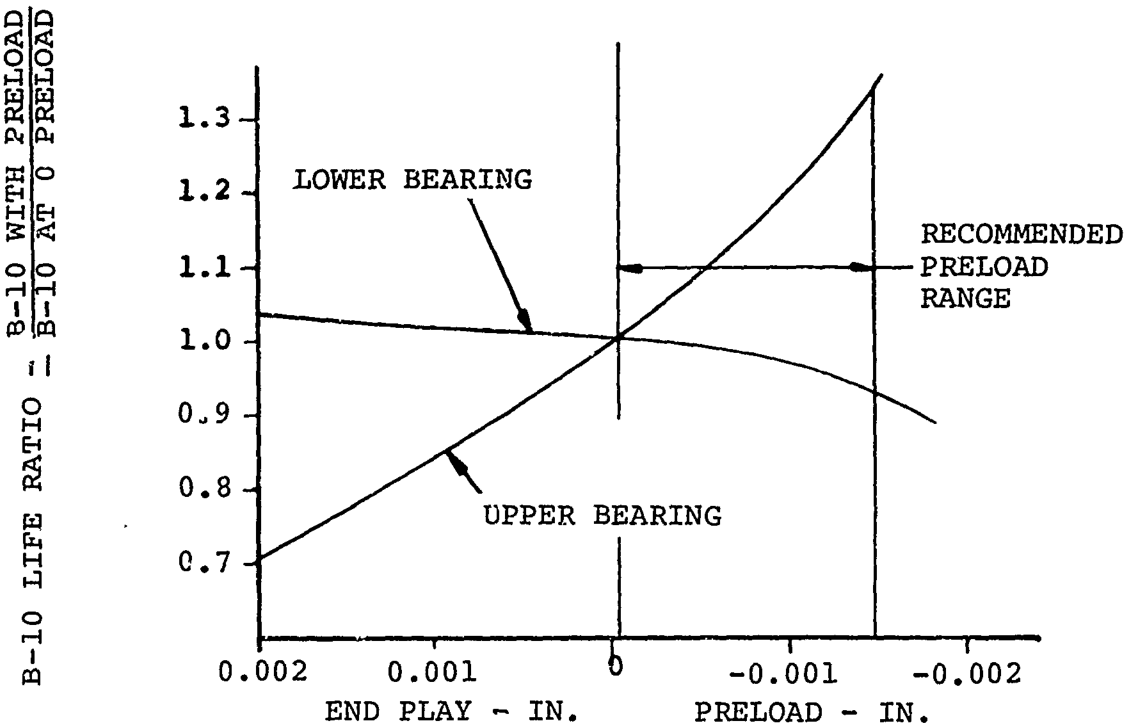
Before full load was applied to the test rig, several hours of running was accumulated with only preload. Bearing performance such as oil return temperature and bearing "g" level was monitored during this period to insure satisfactory bearing operation under recommended preload. After 2 hours of operation, bearing loads were increased until full 100 percent loads were applied. If one set of bearings failed prior to completion of 1200 hours, the failed set of bearings was to be removed and replaced with the spare set of bearings and the test continued until a total of 1200 hours was accumulated on at least one set of bearings or until a second failure occurred.

Prior to disassembly of the test rig, friction torque and axial play of each tapered roller bearing set were measured and recorded. Data was used to determine the amount of preload lost during operation of these bearings.

BEARING PRELOADING PROCEDURE

An important factor which influenced the operating characteristics of tapered roller bearings is proper bearing preload. A study was conducted using available bearing computer programs to show the influence on bearing life versus preload setting. The results of this study, shown in Figure 9 ,indicate that a small preload setting can result in significant improvement in fatigue life of the upper bearing with a slight decrease in life for the lower bearing. Based on this study, an attempt was made to have the initial preload setting for the rotor shaft bearings set at zero (line-to-line) to -0.0015.

Several methods are available for establishing the preload setting for tapered roller bearings. Two methods were investigated in this program. The first method requires that the two tapered roller bearings be assembled onto the shaft and housing



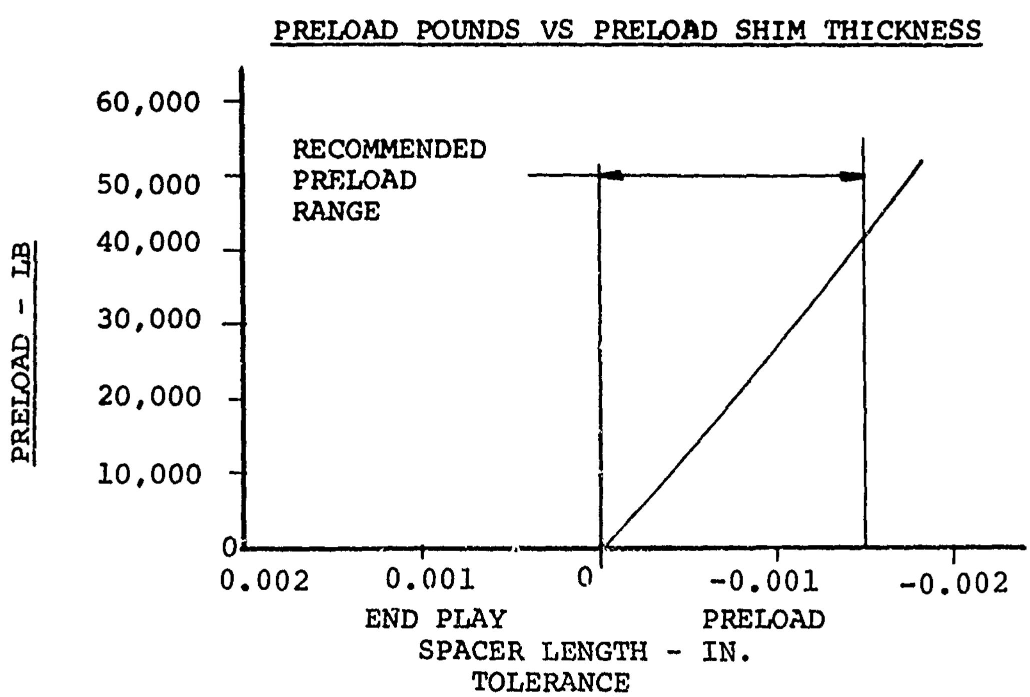


Figure 9. HLH Rotor Shaft Tapered Roller Bearing Preload Study.

with an oversize spacer between the two bearing inner races. The axial movement (end shake) of the shaft with respect to the housing is measured and this amount is then removed from the spacer, plus or minus the preload setting. Because of size, this procedure is very awkward and could result in bearing and shaft damage due to removal of bearings for spacer machining. The setup for this procedure is shown in Figure 10.

The second method investigated in this test program does not require the bearings to be assembled on the shaft for determining spacer dimensions. The procedure for determining the dimension of the spacer to provide the desired preload was:

1. The upper and lower tapered roller bearing cups were installed into the housing assembly.

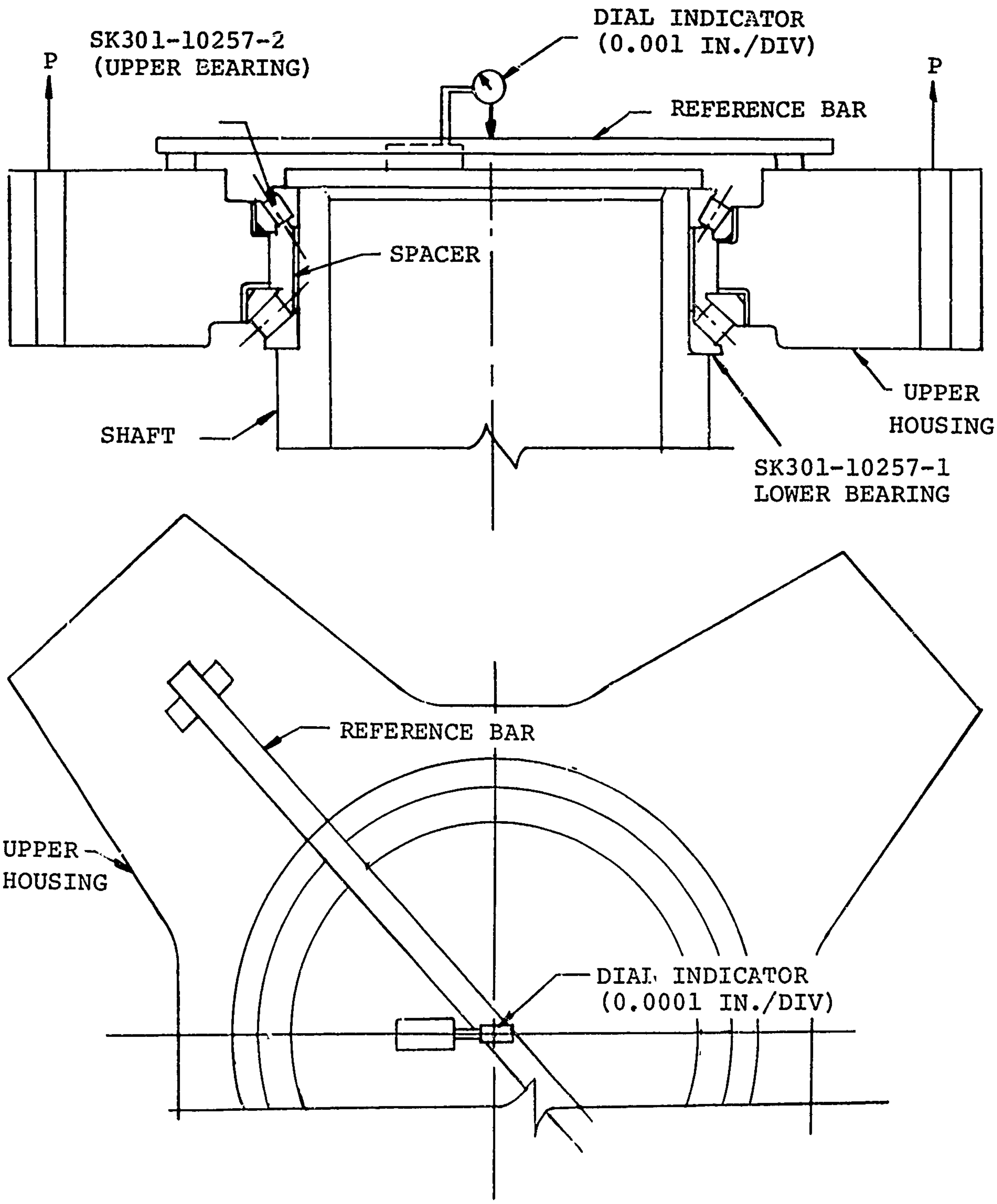
2. The lower bearing cone assembly was placed on a rigid support and the housing assembly placed onto the cone as shown in Figure 11 , sketch A . The bearings were lightly oiled with MIL-L-23699 and rotated in the housing assembly to insure proper seating of the lower bearing assembly. The distance "A" was measured at three places (120° apart). All dimensions were recorded to the fourth decimal place and the average of the three readings used.

3. Item 2 was repeated by turning the housing assembly over and placing the upper bearing cone assembly on the rigid support and measuring distance "B". See Figure 11 , sketch B .

4. After dimensions "A" and "B" were recorded, dimension "C" which is the dimension across the cup front faces (Figure 11 , sketch B) was measured and recorded. The average of three readings was used. The space between the cones in the assembled bearings was then calculated as follows:

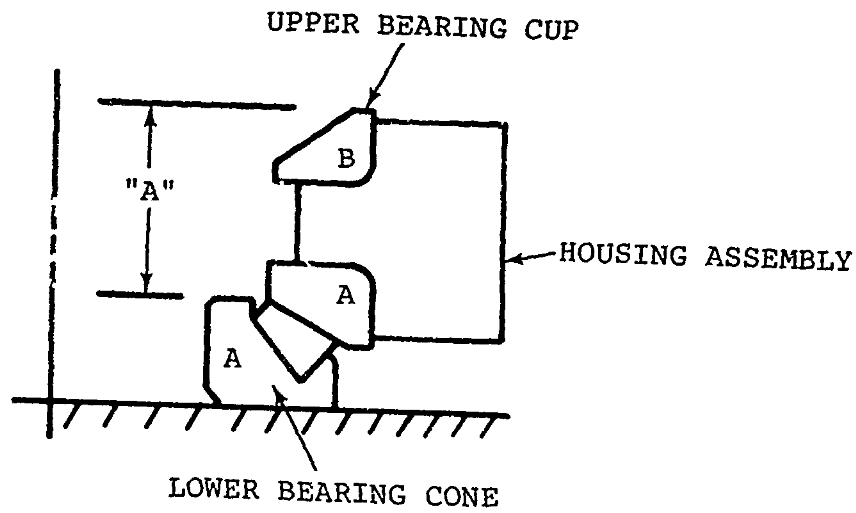
   space between cones = ("A" + "B") - "C"

5. The bore of each bearing and the shaft outer diameter at each bearing seat were measured and recorded to determine the exact shaft fit. Dimensions were recorded to the fourth decimal place. Using all of these dimensions, the following equation was used to determine the size of the spacer to provide the proper preload. The preload spacer length tolerance (P) based on Figure 9 was +0.000 and -0.0015 inch.

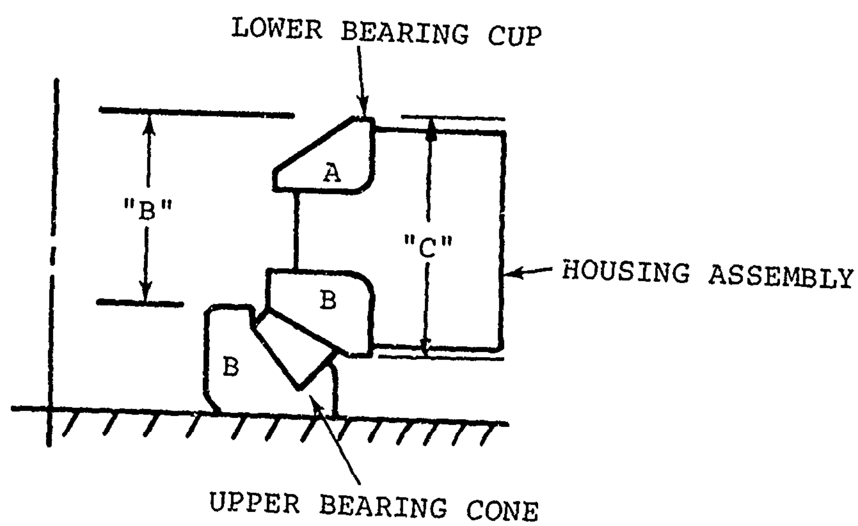


NOT TO SCALE

Figure 10. "End Shake" Determination.



SKETCH A



SKETCH B

Figure 11. Preload Spacer Determined by Analytical Method.

spacer length - "A" + "B" - "C" = $(X_{BRG\ A} + X_{BRG\ B}) + P$

$$X = \text{axial gap removed by shaft fit per bearing} = I \times \frac{b}{c} \times \frac{K}{0.39} \times \frac{1}{2} \times \left[\frac{1 - \left[\frac{a}{b}\right]^2}{1 - \left[\frac{a}{c}\right]^2}\right]$$

where

I - shaft fit - b - (shaft O.D.)

K = tapered roller bearing factor lower bearing A = 0.39
upper bearing B = 0.56

a = shaft inner diameter

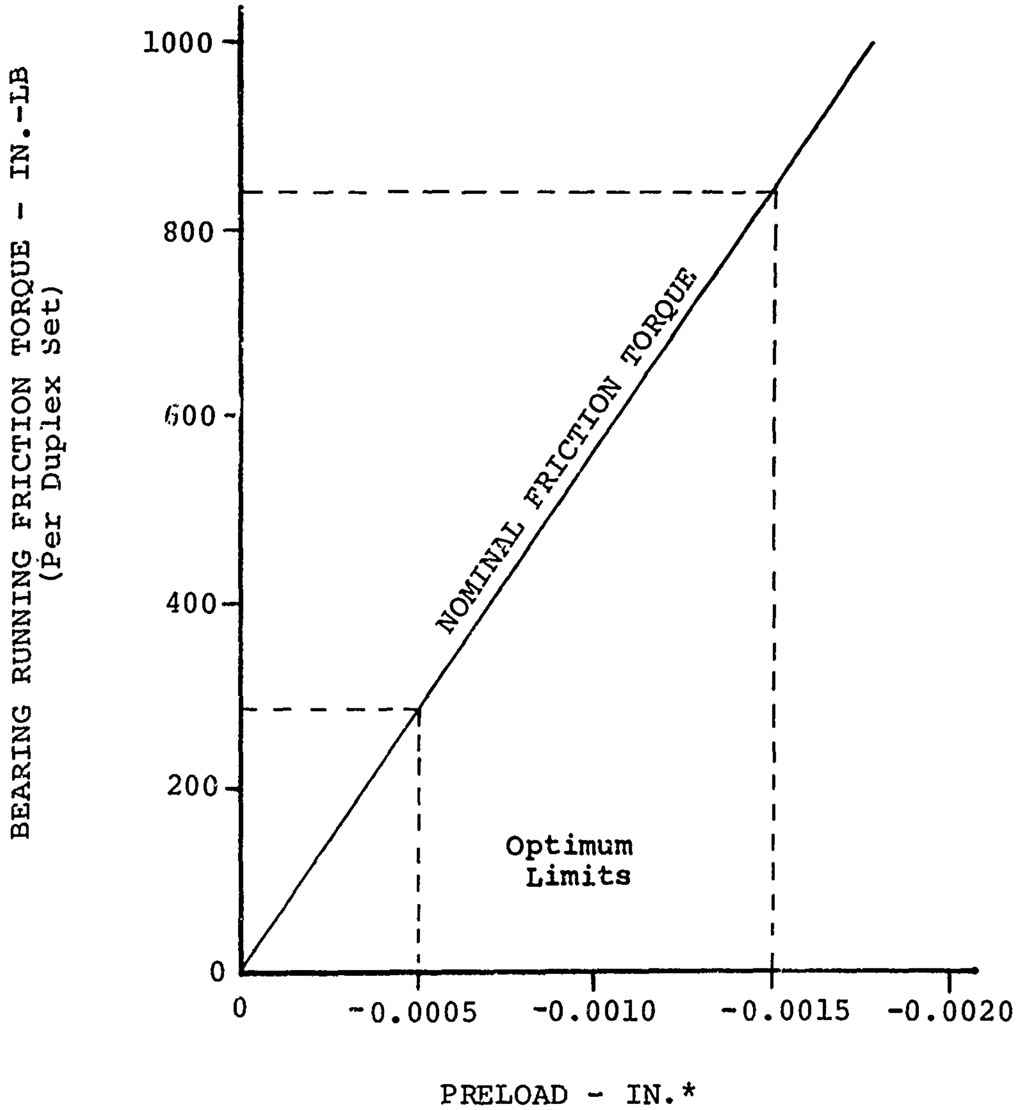
b = bearing bore diameter

c = outer diameter of bearing inner race small end

bearing A = 22.12 in.
bearing B = 21.88 in.

P = preload spacer length tolerance

The bearing spacer was ground to the length dimension established in item 5 and measured prior to assembly of the bearings on the shaft. After assembly, the friction or rotation torque of the set of tapered roller bearings was measured as a check on proper preload. The curve of Figure 12 shows the rotor shaft bearing approximate friction torque for a given preload condition. Verification was attempted of the curve shown in Figure 12 by test data generated in this program.



*Preload to be determined by the difference between the required spacer length and the actual machined spacer length.

Figure 12. Bearing Friction Torque Versus Preload.

TEST DATA AND RESULTS

RESULTS OF TEST RIG ASSEMBLY NUMBER 1

The initial assembly of the test rig consisted of the bearing components on Table V.

TABLE V. IDENTIFICATION OF BEARING COMPONENTS USED IN TEST RIG ASSEMBLY NUMBER 1

Housing Assembly	Bearing Position	Boeing Vertol Part Number	Bearing Serial Number Cup	Bearing Serial Number Cone
Upper	Upper	SK301-10257-2	72-1	72-1
	Lower	-1	72-5	72-1
Lower	Upper	-2	72-3	72-3
	Lower	-1	72-6	72-2

Prior to starting the test, several attempts were made to determine the proper preload spacer length using the analytical method described earlier. The results of this method and preload conditions are summarized in Table VI.

This analytical method resulted in several spacer regrinding operations; i.e., one on the lower housing assembly and four on the upper housing assembly.

A major limitation to this method, as indicated by Table VI, is that there is no assurance that the analytical spacer lengths will be oversized so that they can be reground.

The preloading method which proved much more convenient, simple, and successful in spacer sizing was the experimental method described earlier. This method employed a dial indicator graduated in 0.0001-inch increments and a reference bar to measure the relative displacement between the bearing housing and shaft when the entire assembly was suspended from above as shown in Figure 10 . The dial indicator was mounted on the shaft assembly at its center and the reference bar mounted on the housing. When the assembly was suspended, the dial indicator read the amount of "end shake" in the bearing assembly and therefore the amount to be removed from the preload spacer to eliminate the "end shake". The amount of preload was then specified as a tolerance on the final spacer dimension. The use of this method required only one regrinding operation. This method was used in all subsequent assembly operations.

After sizing the spacers, the bearings and housings were assembled and housing friction torque measured by determining

TABLE VI. AFT TRANSMISSION ROTOR SHAFT TEST BEARING PRELOAD SPACER SIZE DETERMINATION

		Upper Bearing Housing ST50864	Lower Bearing Housing ST50864
Bearing Identification P/N & S/N	Upper Bearing SK301-10257-2	Cup 72-1 Cone 72-1	Cup 72-3 Cone 72-3
	Lower Bearing SK301-10257-1	Cup 72-5 Cone 72-1	Cup 72-6 Cone 72-2
Bearing Cup Outside Diameter	Upper Bearing 25.7874 $^{+0.000}_{-0.003}$ *	25.78462 in. 25.78375	25.78563 in. 25.78579
	Lower Bearing 27.0079 $^{+0.000}_{-0.003}$	27.00631 27.00626	27.00624 27.00619
Bearing Cone Inside Diameter	Upper Bearing 21.1024 $^{+0.000}_{-0.002}$	21.10235 21.10235	21.10112 21.10094
	Lower Bearing 21.1811 $^{+0.000}_{-0.002}$	21.18073 21.18090	21.18027 21.18037
Bearing Housing Seat Inside Diameter	Upper Bearing 25.7774 $^{+0.0030}_{-0.0000}$	25.7808	25.7806
	Lower Bearing 26.9979 $^{+0.0030}_{-0.0000}$	26.9998	26.9992
Shaft Bearing Seat Outside Diameter	Upper Bearing 21.1104 $^{+0.0020}_{-0.0000}$	21.1113	21.1114
	Lower Bearing 21.1891 $^{+0.0020}_{-0.0000}$	21.1899	21.1903
Shaft Bore (18.12 ± 0.02)		18.10	18.10
Axial Gap Removed By Shaft Fit Per Brg. $X=\frac{I}{2}\,\frac{b}{c}\,\frac{K}{0.39}\left[\frac{1-[a/b]^2}{1-[a/c]^2}\right]$	Upper Bearing	0.0050	0.0058
	Lower Bearing	0.0034	0.0037
A	See Sketches A&B, Figure 11	5.2180	5.2172
B		5.1279	5.1341
C		6.1056	6.1143
Space Between Cones = A+B − C		4.2403	4.2370
Spacer Length $= A + B - C + X_{Brg} + X_{Brg}\ B + P$		4.2487 $^{+0.0000}_{-0.0015}$ ** (4.2420)	4.2465 $^{+0.0000}_{-0.0015}$ ** (4.2481)

a = Shaft inner diameter

b = Bearing bore diameter

c = Outer diameter of bearing inner race small end

Lower Bearing A = 22.12 in.
Upper Bearing B = 21.88 in.

I = Shaft fit = b − (shaft O.D.)

K = tapered roller bearing factor

Lower Bearing A = 0.39
Upper Bearing B = 0.56

P = Preload spacer length tolerance = $^{+0.0000}_{-0.0015}$

(*) Dimensions given in parentheses are required values.

(**) Actual Final Spacer Length

the torque required to rotate the bearing housing assemblies on the test bearings. The resulting housing assembly friction running torque was within the limits shown in Figure 12. As indicated in Table VII, the friction torque condition of the upper housing assembly was the zero "end play" (zero preload) condition for the bearing set.

Prior to beginning the test under the prescribed operating conditions, the bearings were run under a "no load" condition without preheating the lubricating oil in order to determine the temperature rise due to the assembly preload heat generated by the bearing friction. The results of this check indicated a 20°F temperature rise as determined by the temperature of the outlet oil from both bearing assemblies. The inlet oil and ambient temperatures were 80°F. A similar check was made with the bearings subjected to the prescribed test loads without preheating the inlet oil. The results of this check indicated a temperature rise of 40°F in the returning oil with oil inlet and ambient temperatures at 80°F.

When the oil inlet temperature was increased to 190°F and testing continued under the prescribed conditions, the oil return temperature was only approximately 175°F. This indicated that the massive housing and shaft were acting as a heat sink and were removing heat from the bearings. Total lubricant flow to each set of tapered roller bearings was 3 gpm. The peak "g" levels recorded during normal operations were between 1.0-2.0g). The test assembly power consumption was approximately 1.5 kw under steady-state conditions.

After completion of the preliminary checks, testing was begun under the prescribed operating conditions (see Table IV). After completing 47.0 hours of total test time, the test fixture was automatically shut down by the fail-safe instrumentation ("vibraswitch", indicating excessive fixture vibration). Investigation of the shutdown revealed that 15 of the bolts used to attach the retainer plate to the shaft (this plate applies the preload clamp-up force to the bearing sets) had failed (Figures 4 and 13), causing the bearings in the upper housing assembly to lose their preload. One of the bolts had failed directly under the head while 14 others failed in the threaded area. This loss in preload, while still under the test thrust and moment input loads, caused the upper bearing in the upper housing assembly to spin on the shaft, severely damaging the shaft bearing seat (Figure 14) and test specimen cone bore (Figure 15). In addition, the lower bearing cone of the upper housing assembly was chipped in the area where it seats on the shoulder provided on the shaft (Figure 16). Damage was due to the retainer in the housing hitting the bearing after the bolts failed. All the remaining test specimens spun

TABLE VII. SUMMARY - TEST BEARING OPERATING CONDITIONS

		Test Run -1		Test Run -2	
		Upper Bearing Housing ST50864	Lower Bearing Housing ST50864	Upper Bearing Housing ST50864	Lower Bearing Housing ST50864
Bearing Identification P/N & S/N	Upper Bearing SK301-10257-2	Cup 72-1 Cone 72-1	Cup 72-3 Cone 72-3	Cup 72-4 [3] Cone 72-4	Cup 72-3 [2] Cone 72-3
	Lower Bearing SK301-10257-1	Cup 72-5 Cone 72-1	Cup 72-6 Cone 72-2	Cup 72-8 [3] Cone 72-4	Cup 72-6 [2] Cone 72-2
Required Spacer Length (in.) - Analytical Method -		4.2487 $^{+0.0000}_{-0.0015}$	4.2465 $^{+0.0000}_{-0.0015}$	-	-
Actual Final Spacer Length (in.) - Empirical Method -		4.2420	4.2481	4.2411	4.2554
Shaft/Cone Interference Fit	Upper Bearing	0.00895	0.01037	0.0134	0.0143
	Lower Bearing	0.00909	0.00998	0.0127	0.0134
Pretest Bearing Set Preload Friction Torque - Breakaway (in.-lb)		126.3 [1]	670.5	-	740.0
Pretest Bearing Set Preload Friction Torque - Running (in.-lb)		108.7 [1]	541.2	-	680
Posttest Bearing Preload Friction Torque - Running (in.-lb)		-	-	-	600
Pretest "End Shake " (in.)		0.0000	0.0000	0.0000	0.0000
Posttest "End Shake" (in.)		-	-	0.0066	0.0000

1 This is zero "end shake", zero preload condition for this bearing set.

2 Bearing Cones I.D. Copper Plated after Test Run 1.

3 Cup O.D. Copper Plated.

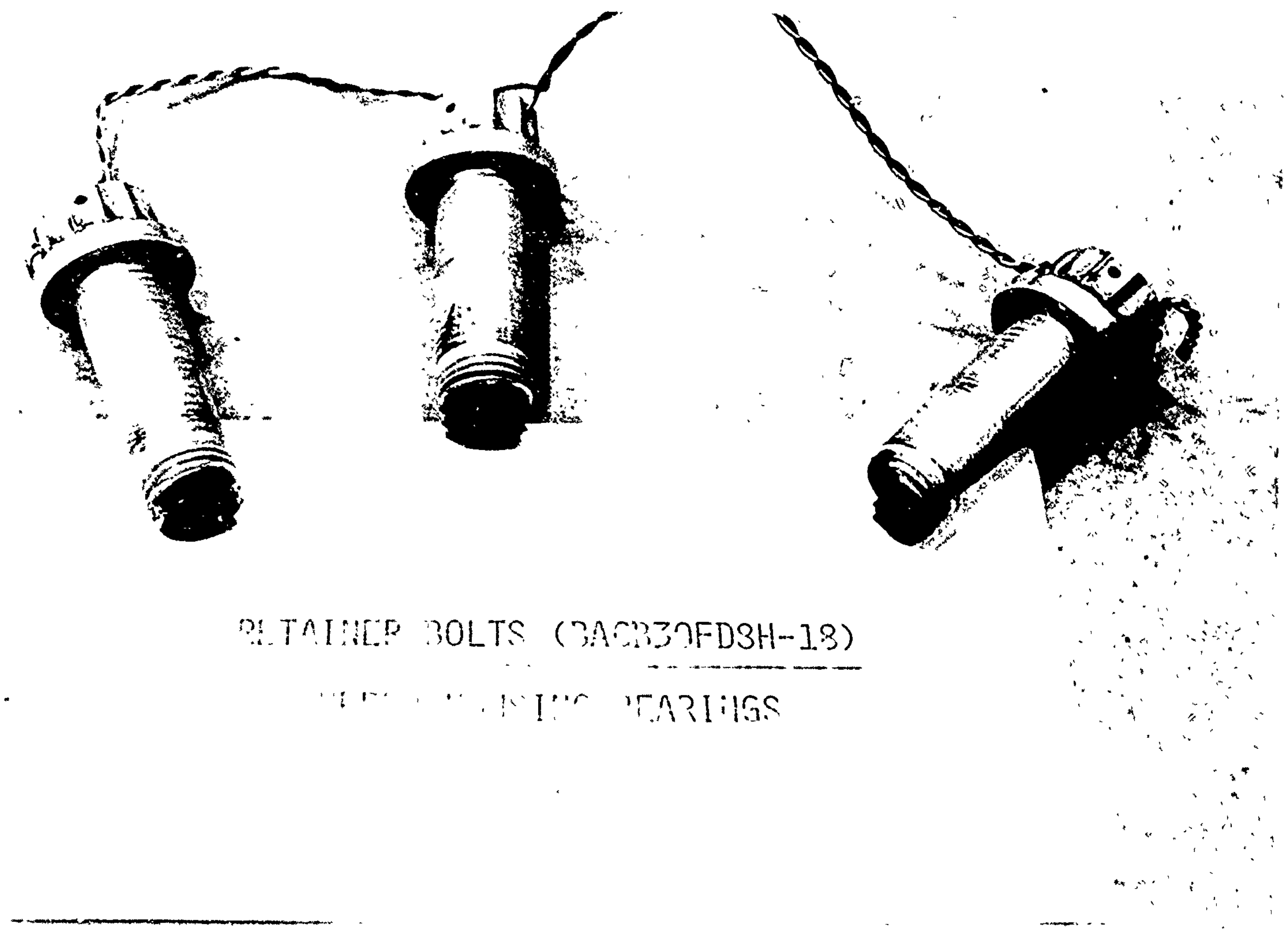


Figure 13. Typical Preload Clamp Plate Bolt Failures.

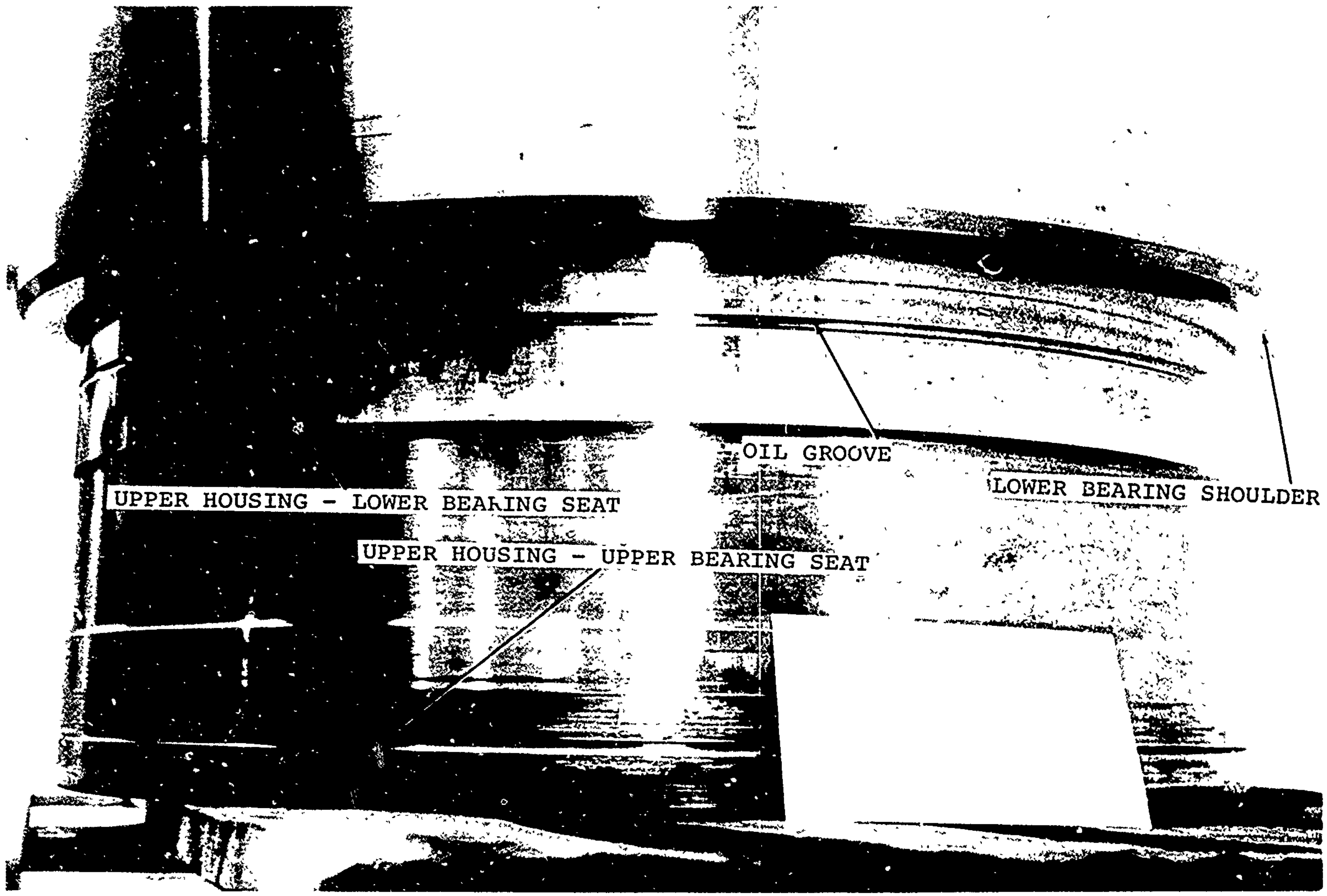


Figure 14. Condition of Shaft of Upper Housing Bearing Assembly.

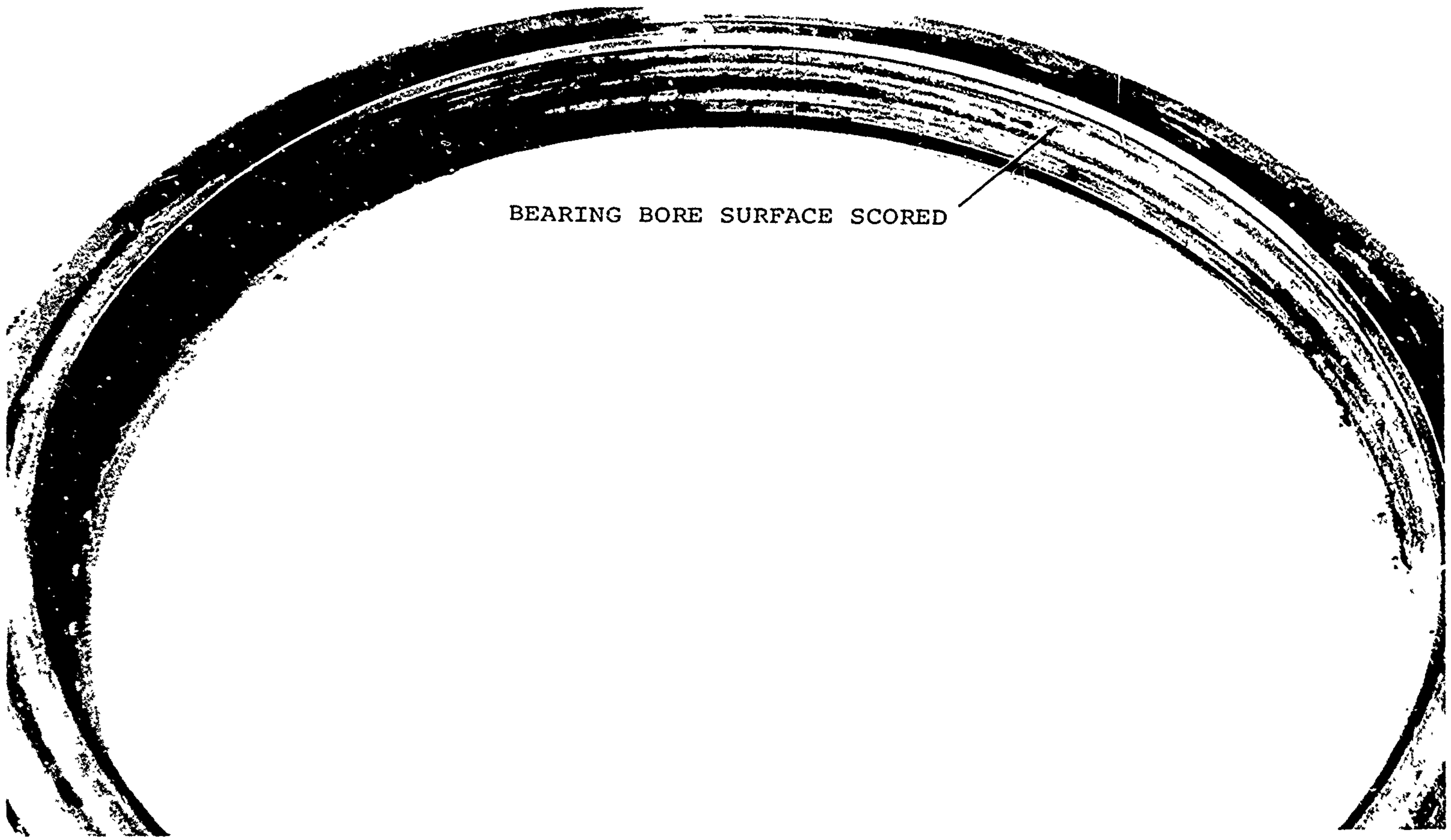


Figure 15. Condition of Bore of SK301-10257-2 Bearing (Upper Housing).

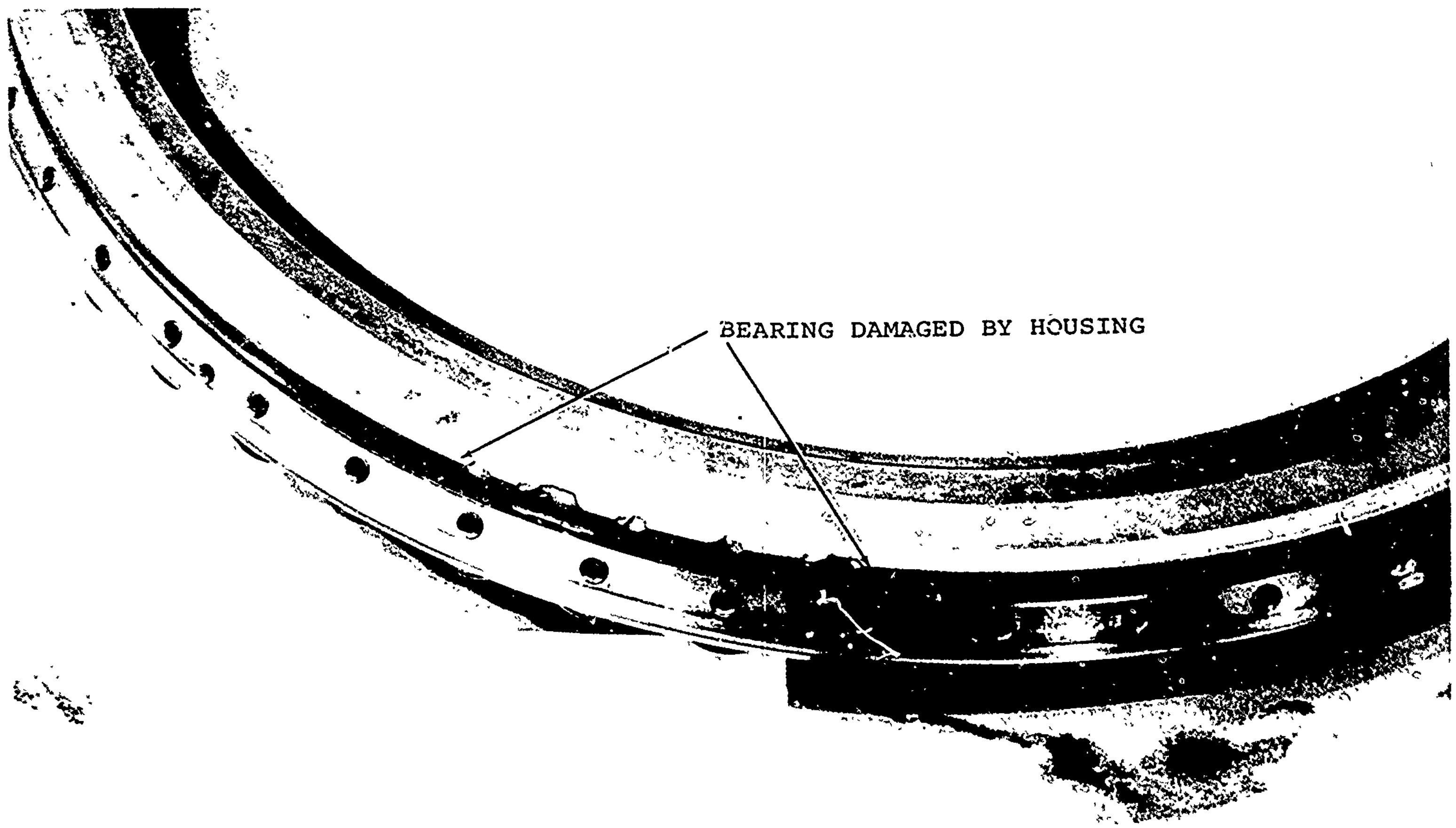


Figure 16. Condition of SK301-10257-1 Bearing (Upper Housing).

on their seats, resulting in extensive damage to the shaft and preload plates (Figures 17, 18, and 19).

The cause of the preload bolt failures was initially attributed to the installation of improper flat (AN960) washers (washers were not countersunk in the bore to provide clearance for the radius underneath the bolt heads) at assembly, causing a bending load to be induced under the head of the bolts which, when combined with the fatigue loads induced by the test conditions, resulted in bolt failure.

In order to repair the damaged shaft areas, the bearing seats were spray-welded with 440C stainless steel (Rc (55-62)). The bearing seat outside diameters were increased so that the allowable interference fit was increased from 0.0080-0.0120 inch to 0.0012-0.0016 inch. In addition, the number of retaining bolts was increased from 16 to 32 in both the upper and lower housing assemblies.

The damaged bearings from the upper housing assembly were returned to the manufacturer for rework, and the preload plates from the upper and lower housing assemblies were resurfaced to remove the galled areas (Figures 18 and 19).

As an added deterrent to excessive galling damage caused by relative motion between the test bearing components and their seats (cones) or housings (cups), the bearing cups in the upper housing assembly were plated with alkaline copper to a thickness of approximately 0.0003 inch prior to the second test buildup.

RESULTS OF TEST RIG ASSEMBLY NUMBER 2

The second assembly of the test rig consisted of the bearing components on Table VIII.

TABLE VIII. IDENTIFICATION OF BEARING COMPONENTS USED IN TEST RIG ASSEMBLY NUMBER 2

Housing Assembly	Bearing Position	Boeing Vertol Part Number	Bearing Serial Number Cup	Cone
Upper	Upper	SK301-10257-2	72-4	72-4
	Lower	SK301-10257-1	72-8	72-4
Lower	Upper	SK301-10257-2	72-3	72-3
	Lower	SK301-10257-1	72-6	72-2

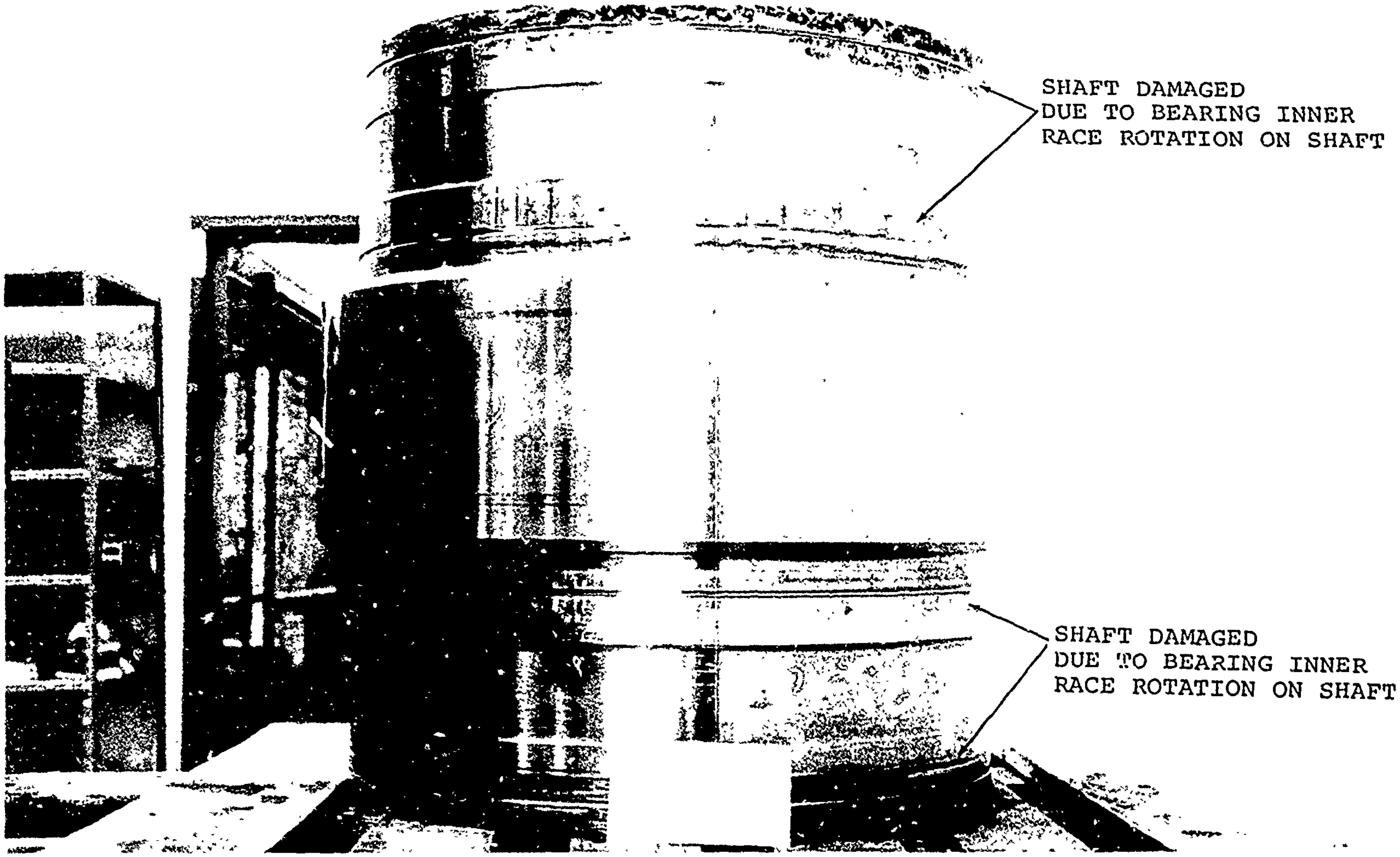


Figure 17. Rotor Shaft Condition After Test No. 1.

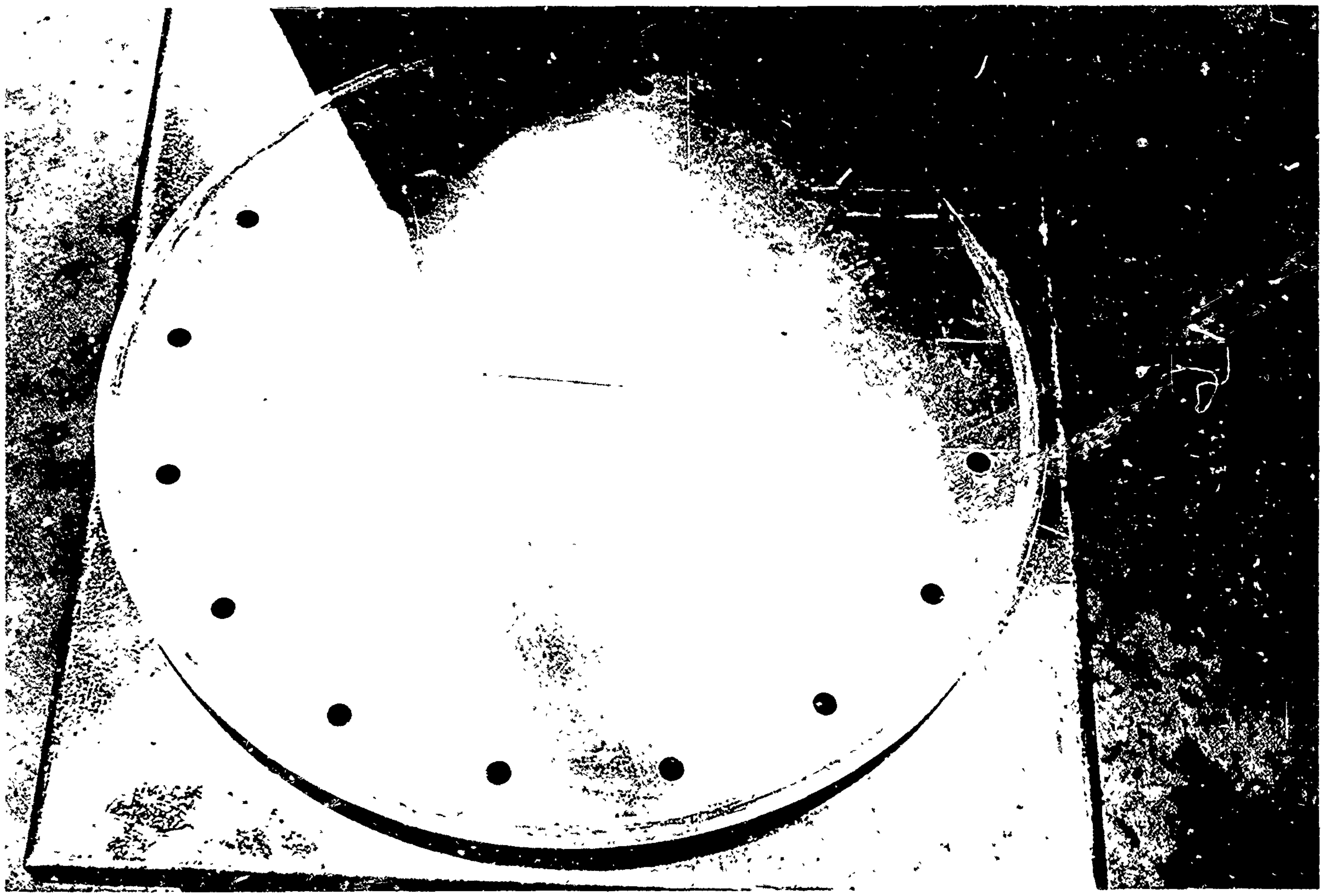


Figure 18. Upper Bearing Assembly Preload Plate Scored due to Bearing Rotation on Shaft.

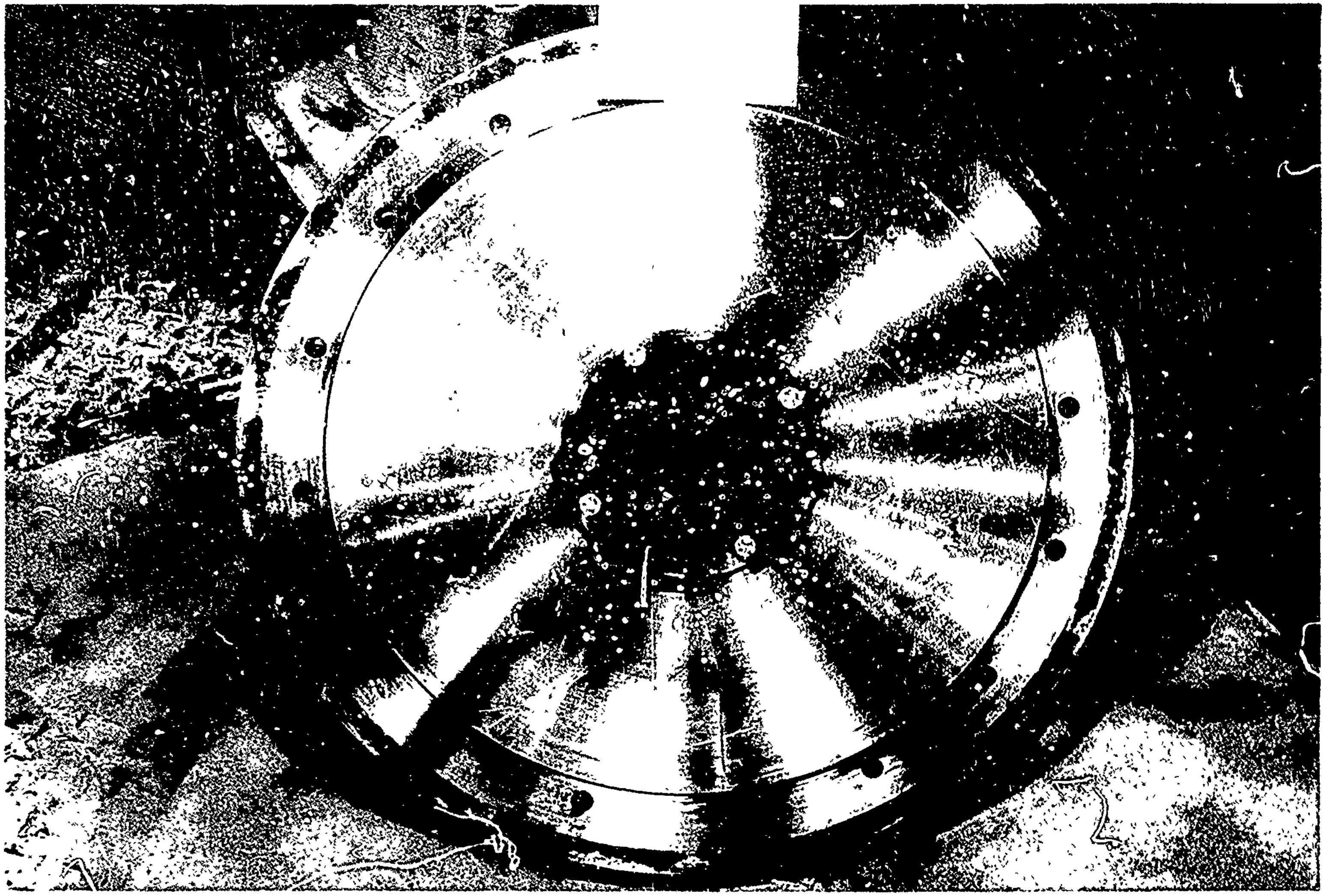


Figure 19. Lower Bearing Assembly Preload Plate Scored due to Bearing Rotation on Shaft.

The bearings comprising the lower housing assembly were the same as for Test 1.

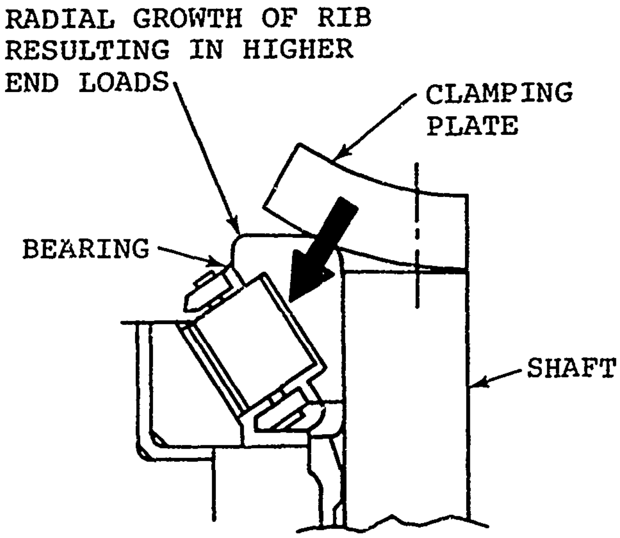
After spray welding the shaft journals, the bearings and housings for the second test were assembled on the shaft to determine the size of the preload spacers. When the bearings were removed to permit grinding of the spacers, the spray-welded stainless steel bearing seats cracked, lifted, and peeled away from the parent shaft material.

After removing the stainless steel bearing seats, the seats were rebuilt by Union Carbide with a tungsten carbide (Union Carbide designation LW-1N40) surface (Rc-72) using a detonation spray process. The seats were precision-ground to final size with a diamond abrasive wheel.

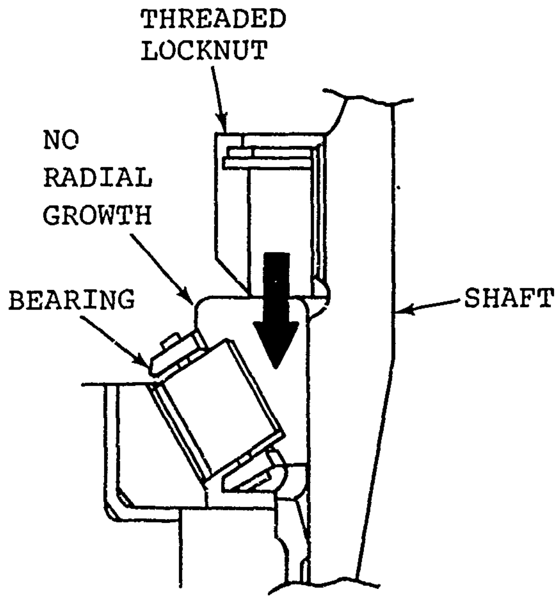
It was noted that the assembly and disassembly of the test specimens on the bearing seats were much easier with the tungsten-carbide seats than with either the original surfaces (9310 steel Rc 34-38) or the stainless steel surfaces. This was attributed to two factors: first, experience gained in the assembly process and second, the increased hardness of the tungsten-carbide surface and the superior surface finish; i.e., 8-12 inches rms with the tungsten carbide as opposed to 32 inches rms with the 9310 steel and 440C stainless steel surfaces. The initial conditions of the test assembly before starting the second test run are given in Table VII.

Testing under the second run was conducted for a total of 1093 hours. A cross-check of the test parameters indicated that the initial 400 hours of testing under the second run was run at a 57.50×10^3 pounds thrust load and a 1.96×10^3 inch-pound moment instead of 70.80×10^3 pounds thrust and 2.16×10^3 inch-pound moment. The effect on life of incorrect loads will be discussed in a later section.

During testing under this phase, there were 10 instances where the preload bolts in the upper assembly failed, causing the test fixture to shut down as a result of an increased vibration level. An inspection and analysis of the bolt failures revealed that they were fatigued by excessive bending of the bearing preload plate. This plate bending caused nonuniform contact of the bolt heads, causing a bending load to be introduced which resulted in the early failure of the bolts. A further attempt to solve this problem was made by introducing a spacer ring between the preload plate and the top of the drive shaft (Figure 4) to eliminate bending in the area of the bolts. These spacers had only minimal success. In addition, the plate bending produced a radial load on the upper bearing cone rib (Figure 20) which may have affected the true preload setting and also produced higher loads on the roller ends. This loading would not occur if a locknut was used to retain the



TEST FIXTURE
INSTALLATION



TRANSMISSION
INSTALLATION

Figure 20. Effect of Preload Plate Bending on Upper Bearing Cone Rib Deflection.

bearings. Of the 10 instances of bolt failures in the second test series, there were 2 instances of preload bolt failure in the lower assembly. This relative infrequency was due to the greater stiffness of the lower preload plate and the use of a pilot on the plate which reduced the plate bending underneath the bolt heads (Figure 4).

After 899.7 hours of testing under test run 2, the audible noise level of the test assembly increased appreciably and continued to do so until testing was suspended at the end of 1093.0 hours. This was the first indication of a possible specimen failure. There were 5 machine shutdowns to change preload bolts between the first indication of a possible specimen failure and test termination.

An evaluation of the test hardware after the test suspension revealed that the lower bearing in the upper assembly had failed. Thirty-eight of the rollers had spalled as well as the cup in approximately 6 areas. The cup failed in the area of highest load application (areas R1 in Figure 7) which was also the area of greatest housing stiffness. Figures 21, 22, 23, and 24 show the failed components.

The bearing failure may have been influenced by the shock loads applied as a result of the frequent loss of preload when the preload bolts failed and edge loading of the rollers resulting from a lateral growth of the bearing cone due to radial loading caused by the preload plate deflection. Prior to disassembly, the final end shake and rotational friction torque conditions of the assemblies were determined. These values are shown in Table VII. Although the bearings spun on their shaft seats during the last series of tests, the tungsten-carbide surfaces did not exhibit areas of galling or fretting. The shaft had a highly polished area on the seat of the upper bearing in the upper housing assembly. Figures 25, 26, and 27 show the bearing seats after testing. It also appeared that the copper plating had a beneficial effect in that the bearing cones in the lower housing assembly had not been scored due to rotation on their shaft seats.

During disassembly, the forces required to remove the bearings from their seats were recorded. With hydraulic oil at approximately 2000 psig supplied to the oil groove, the force required to slide the bearings off the shaft was 24 x 10 pounds.

INSPECTION OF TEST BEARINGS

After completion of all testing, the 2 bearings (SK 301-10257-1) which were in the test rig for the final 1093 hours were returned to the Timken Company for their detailed inspection and metallurgical analysis. The 2 bearings (SK 301-10257-2) were

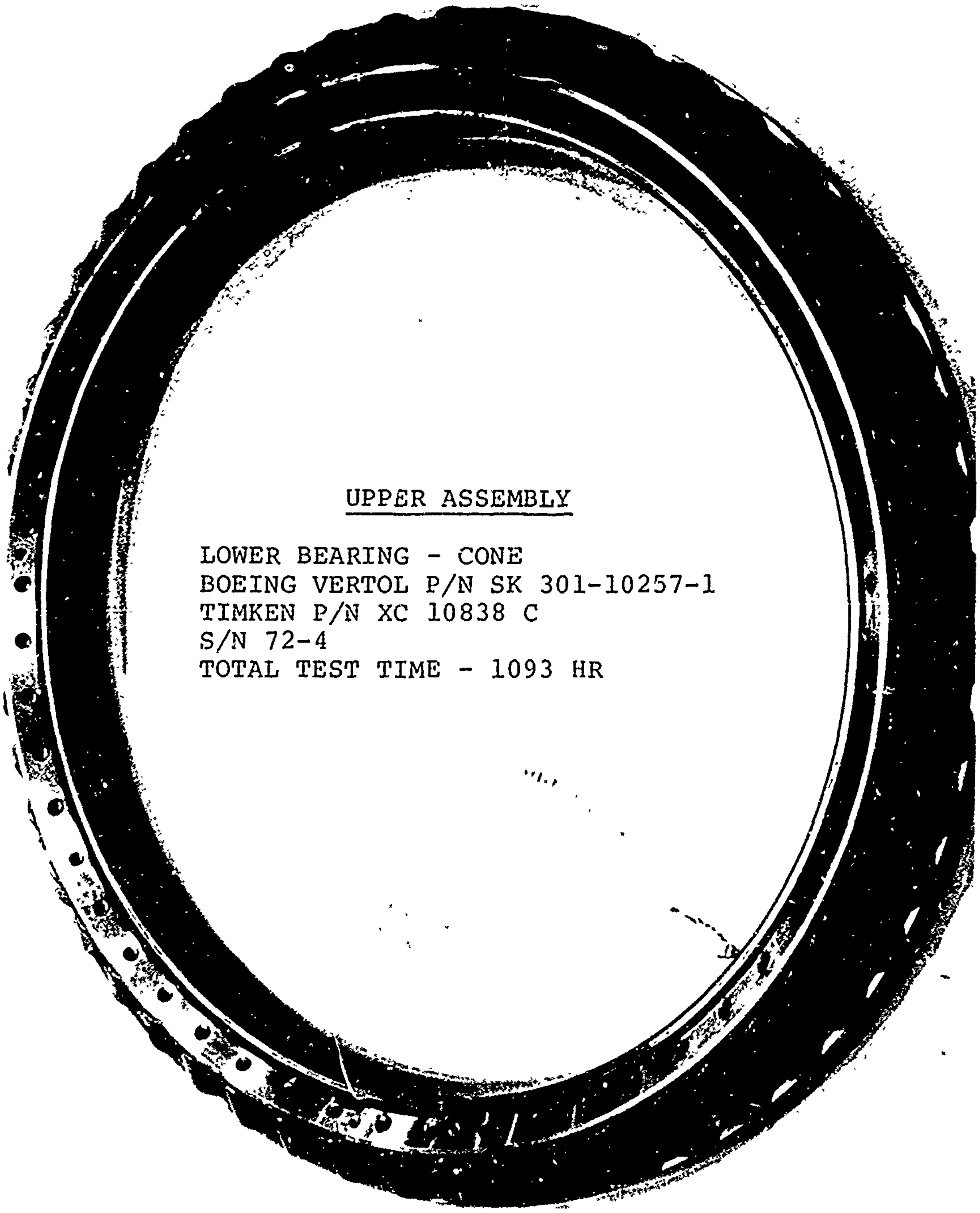


Figure 21. Failed SK301-10257-1 Cone Assembly.

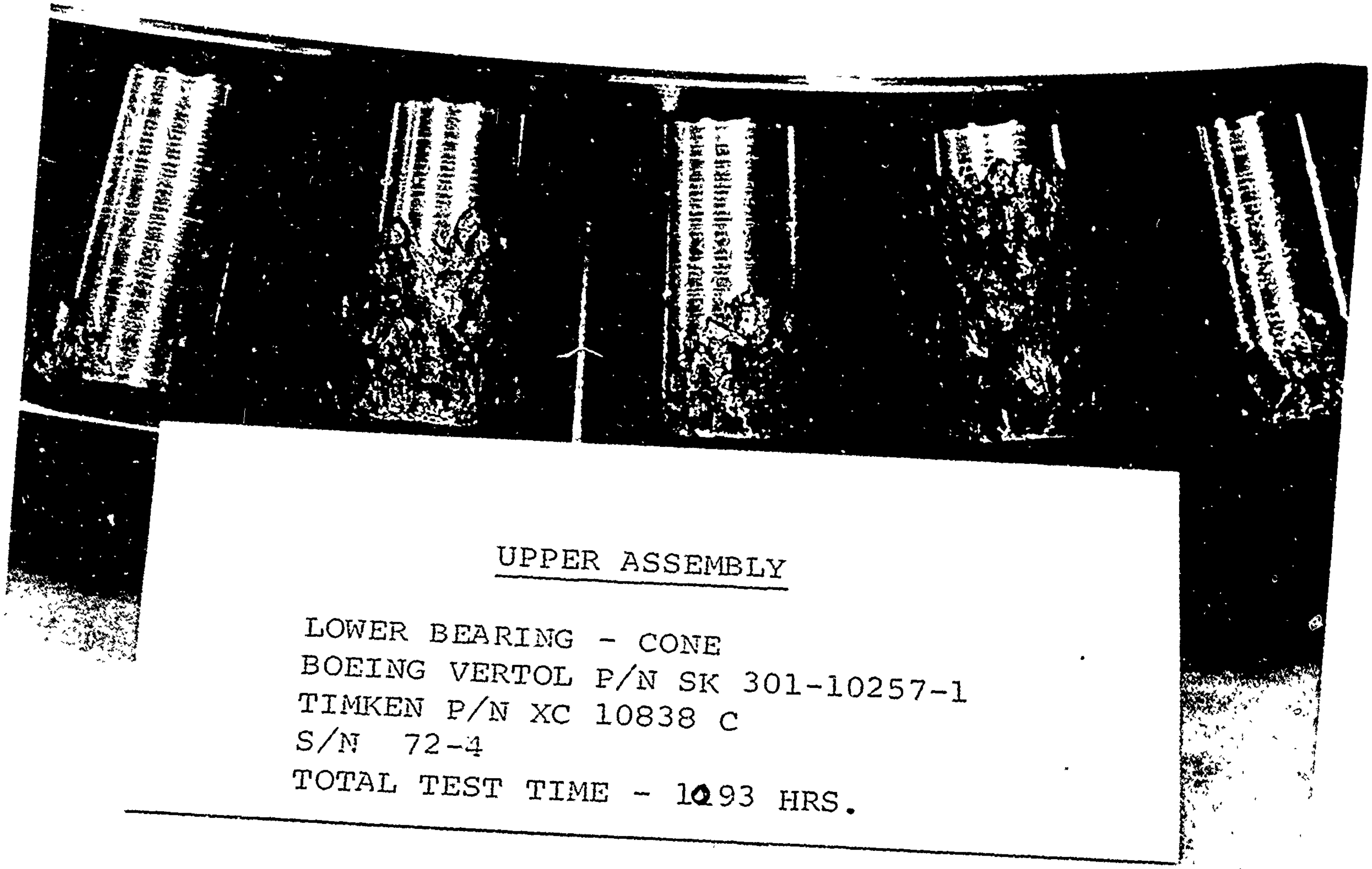


Figure 22. Failed Rollers of SK301-10257-1 Bearing.

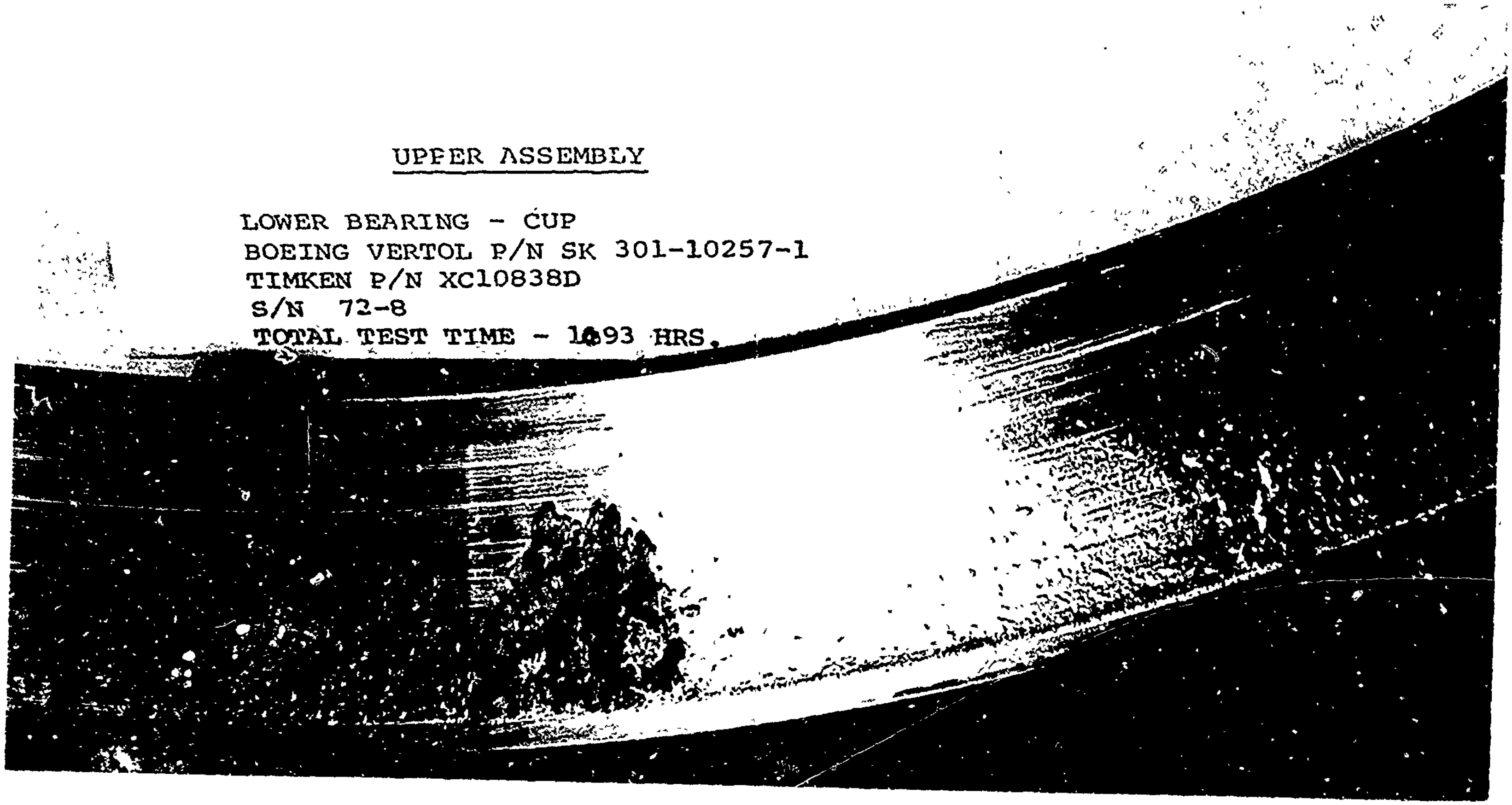


Figure 23. Spalled Cup of SK301-10257-1 Bearing.

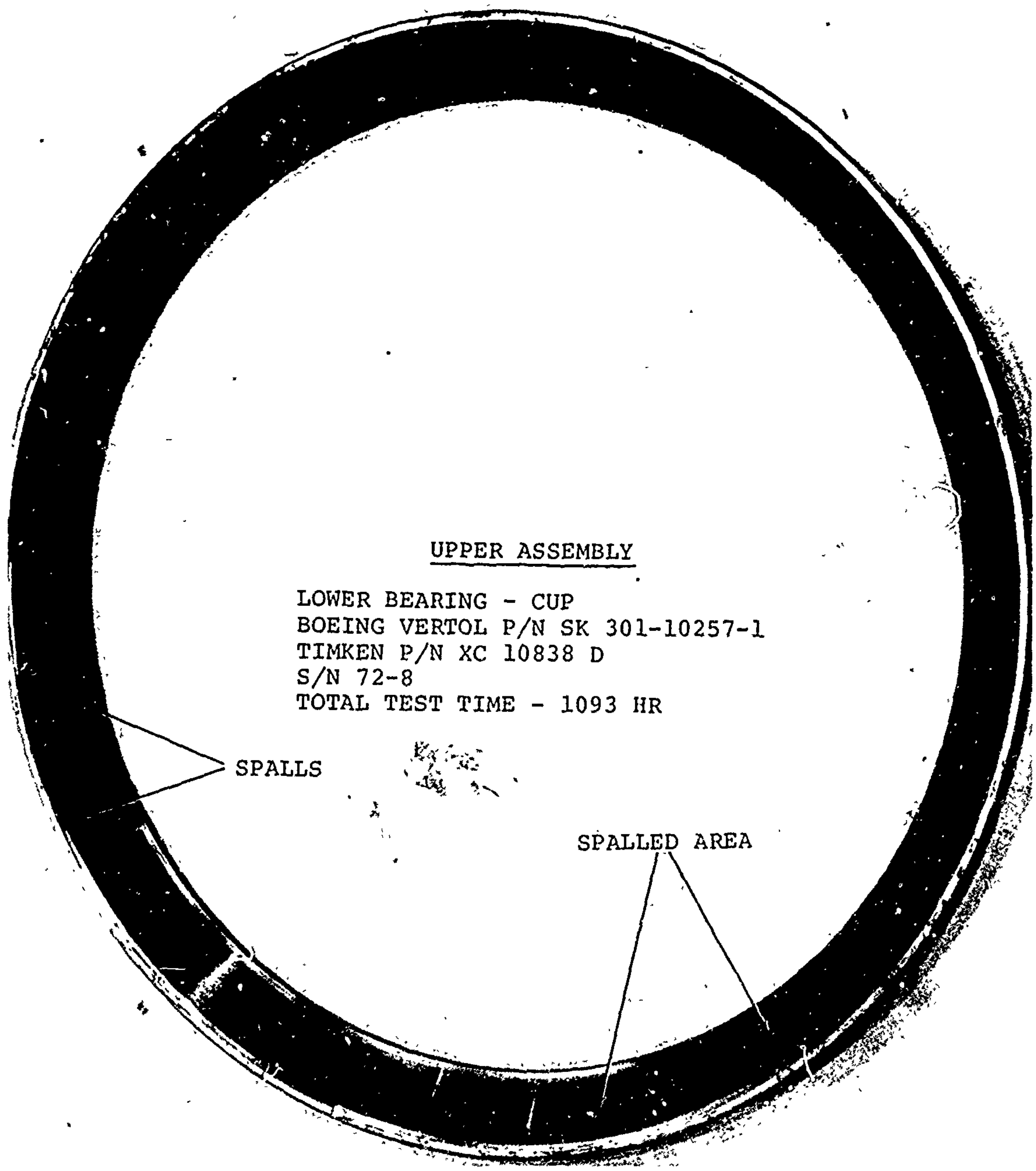


Figure 24. Location of Spalled Areas on Cup of SK301-10257-1 Bearing.

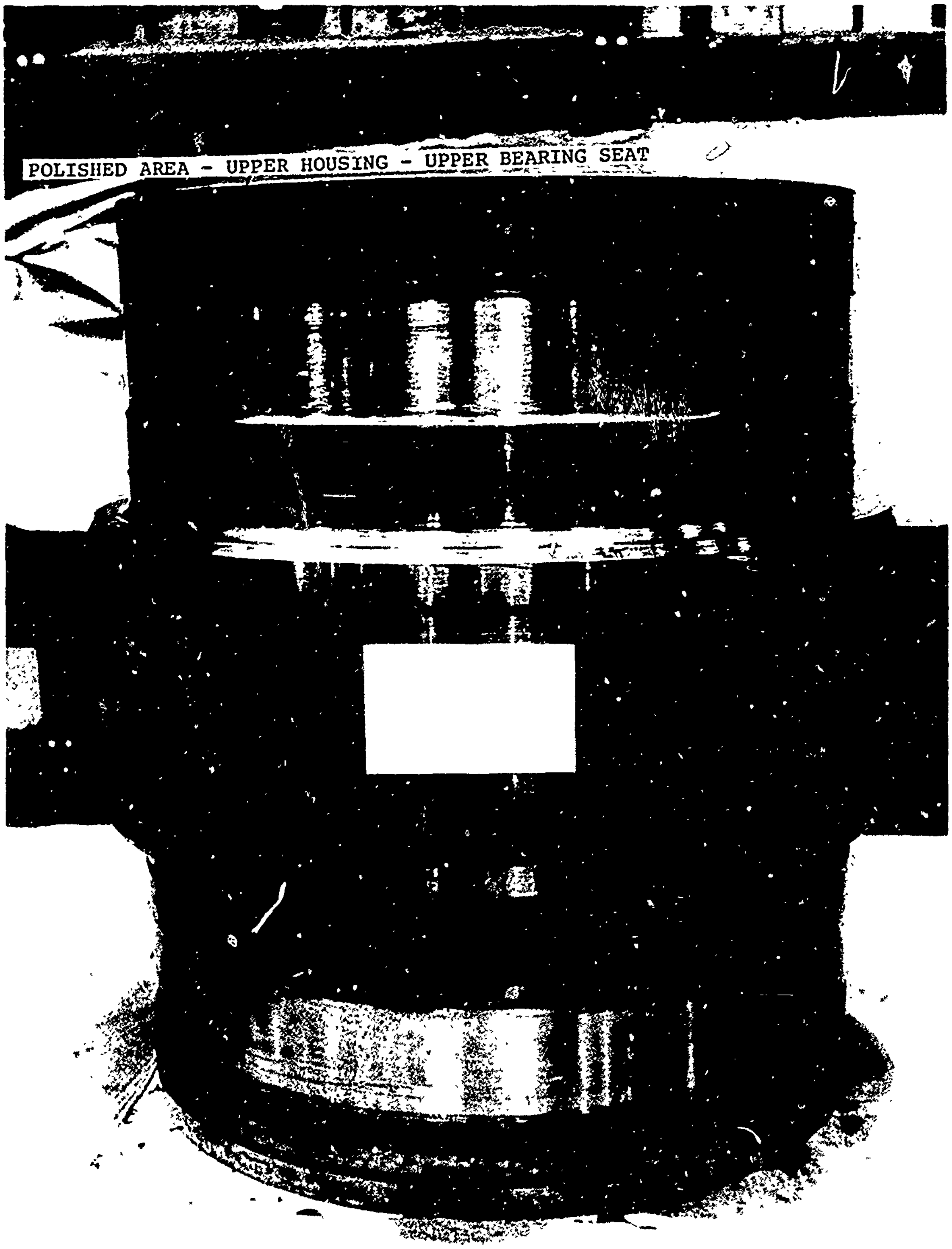


Figure 25. Rotor Shaft Condition After Test 22.

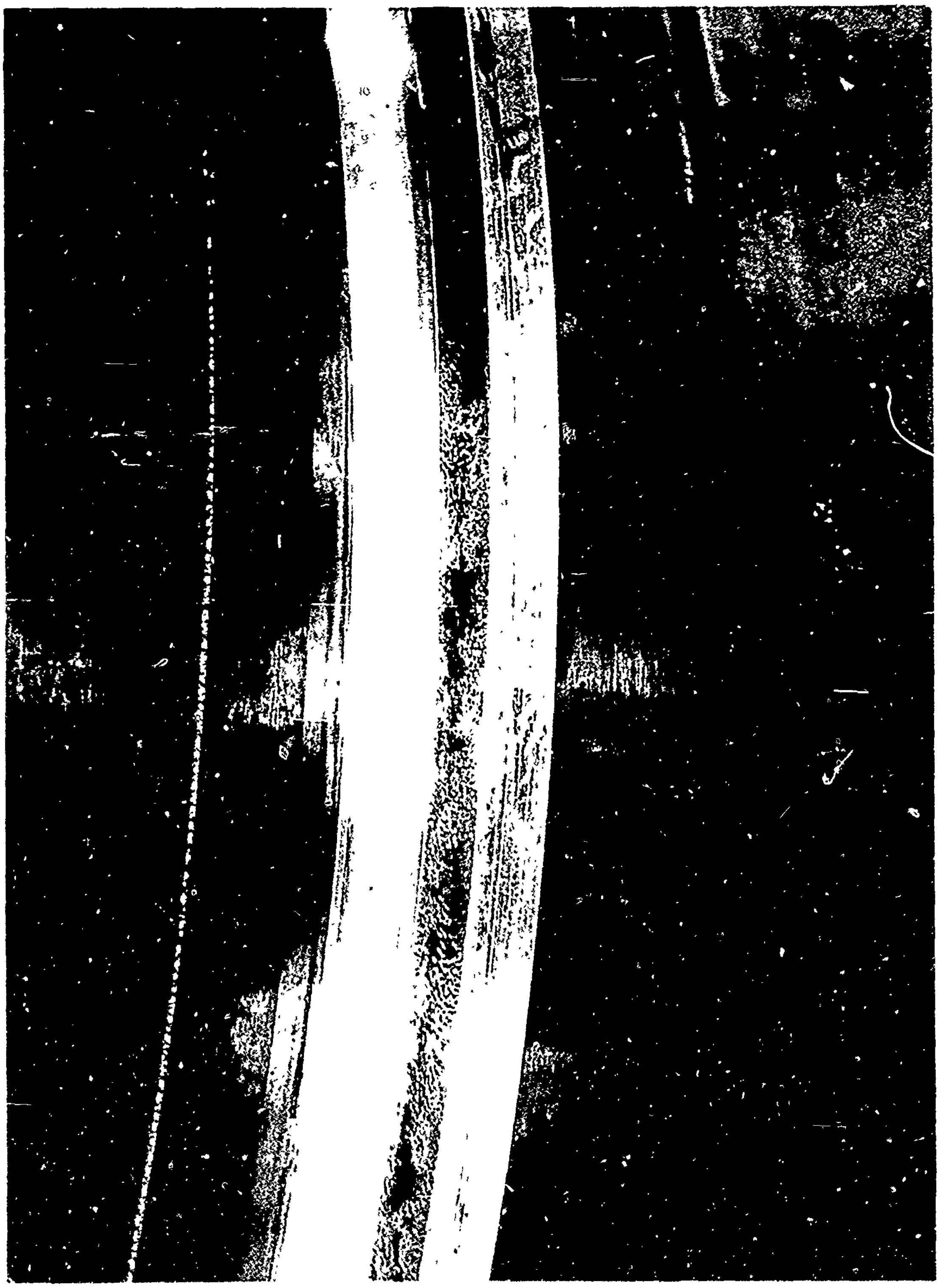


Figure 26. Tungsten-Carbide Bearing Seat - Upper Housing, Upper Bearing.

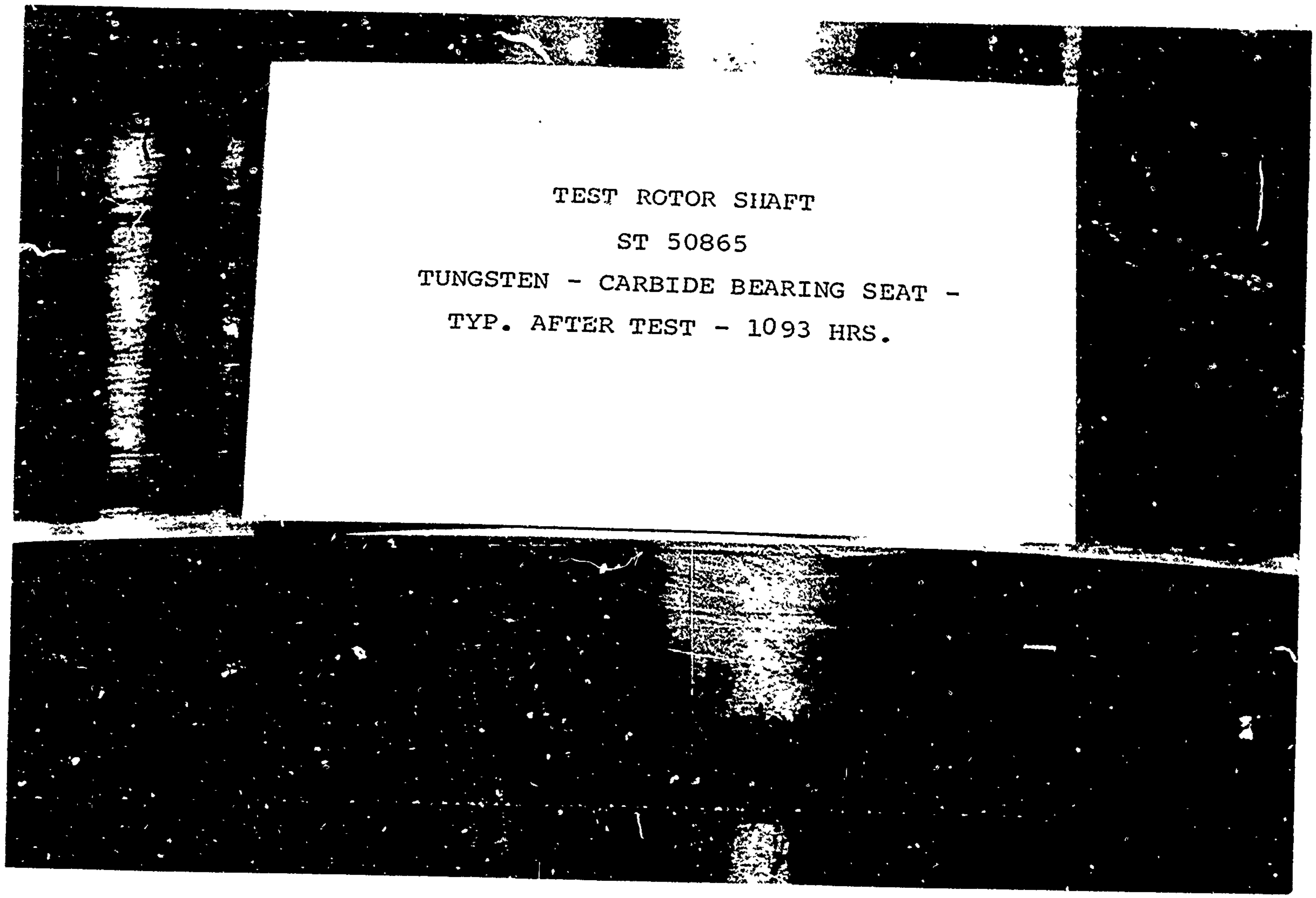


Figure 27. Tungsten-Carbide Bearing Seat Lower Housing.

visually examined at Boeing Vertol and were found to be free of any damage; therefore, these bearings were not returned to Timken for any additional inspection. The results of the Timken investigation are summarized below.

One bearing assembly (cone S/N 72-2 and cup S/N 72-6) was found to be free of an evidence of fatigue damage, while the second bearing assembly (cone S/N 72-4 and cup S/N 72-8) had fatigue damage at the large ends of the cone, cup, and rollers. The results of the visual inspection of both bearings are summarized in Table IX.

A visual inspection of the cones, rollers, and cage of bearing S/N 72-2 showed evidence of etch staining due to leakage of copper plate solution during plating of the bore and back-face area. Similiar etching from the copper plating solution was also observed on the cup surface of S/N 72-8. This type of damage would not have occurred if the bearing bore, outer diameter, and back-face surfaces were plated prior to final assembly of the bearing. A review of the copper-plated and non-copper-plated surfaces showed that the copper plate did minimize the fretting of the surfaces.

Cup S/N 72-8 showed heavy bruising and geometric stress concentration type spalling on the raceway. The two areas of geometric stress concentration spalling appeared in two areas 90° apart. The areas were associated with the two arms of the housing applying the compressive link loads. One spall, 4 inches long, extended approximately three-fourths of the way across the race. The other spall was somewhat larger (6 inches long), but extended only one-third of the way across the race (Figure 30).

Thirty-eight rollers showed typical geometric stress concentration spalling on the large end, in some cases extending two-thirds of the way down the roller. Some rollers also showed irregular cracking across the large end extending from the radius blend to the cage pin hole (Figure 34).

Transverse sections were cut through the large ends of the rollers containing the transverse cracking. Microscopic examination showed the cracks to extend from the outside radius blend of the roller and into the cage pin. Further examination of the cracks showed no inclusion material, scale or anything which would indicate that their source was caused by something besides the stresses caused by the geometric stress concentration spalling.

A metallurgical analysis was conducted on the components of the second bearing assembly (cone S/N 72-4 and cup S/N 72-8).

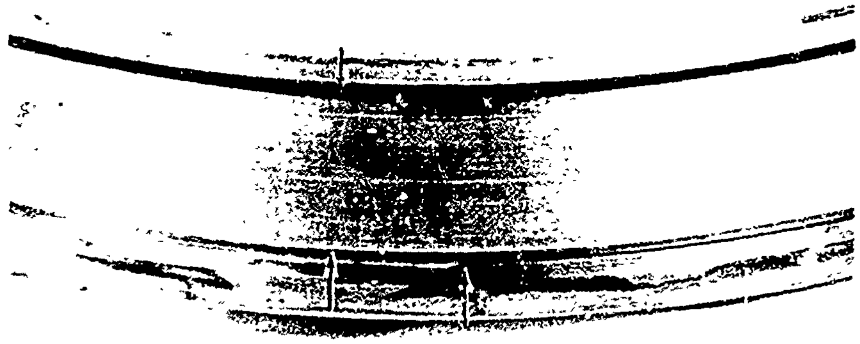
TABLE IX. SUMMARY OF VISUAL EXAMINATION OF TEST BEARINGS

Component / Bearing Location On Test Rig	Upper Bearings Upper Housing Cup S/N 72-4 Cone S/N 72-4	Lower Bearing Upper Housing Cup S/N 72-8 Cone S/N 72-4	Lower Bearing Lower Housing Cup S/N 72-6 Cone S/N 72-2	Upper Bearing Lower Housing Cup S/N 72-3 Cone S/N 72-3
Cone	No fatigue damage on raceway surface	Two small spalls on race and debris damage (Figure 28) Bore and back face show heavy fretting (Figure 29)	No fatigue damage, random debris brusing and etch marks across race at 3 locations (Figure 28) Copper-plated back face and bore showed some fretting (Figure 29)	No fatigue damage on raceway surface
Cup	No fatigue damage on raceway surface	Some copper plating evident on race Some spalls on large end extending on to race Numerous edge type loading spalls at large end. Spalls concentrated in two areas 90° apart. No fretting of copper plate on O.D. (Figures 30 and 31)	No fatigue damage, light debris bruise marks and false brinell marks on race (Figure 30) Light fretting on O.D. (Figure 31)	No fatigue damage on raceway surface

TABLE IX. Continued

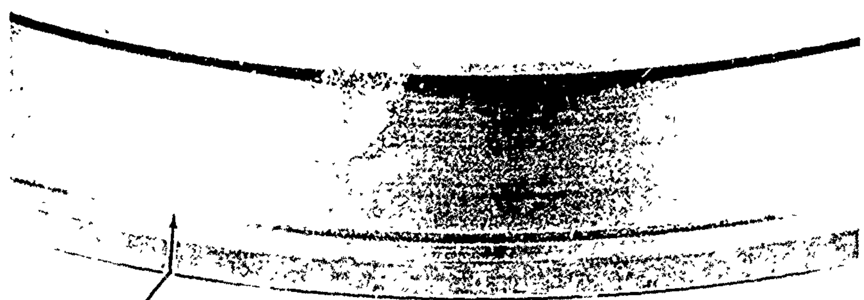
Bearing Location On Test Rig / Component	Upper Bearing Upper Housing Cup S/N 72-4 Cone S/N 72-4	Lower Bearing Upper Housing Cup S/N 72-8 Cone S/N 72-4	Lower Bearing Lower Housing Cup S/N 72-6 Cone S/N 72-2	Upper Bearing Lower Housing Cup S/N 72-8 Cone S/N 72-3
Rollers	No evidence of wear or fatigue.	38 rollers spalled at large end (Figure 32) 5 spalled rollers also had cracks extending from spall onto spherical end and one continued into pin bore (Figure 34)	The O.D. of several rollers have etch marks (Figure 32) No fatigue damage	No evidence of wear or fatigue
Cage	No evidence of wear	No wear measured on pin or end rings (Figure 33)	Heavy copper plate etching noted on end ring surface. No wear measured on pin (Figure 33)	No evidence of wear

CONE NO. 1 SERIAL NUMBERED 72-2



NOTE ETCH MARKS ON RACE & RIB

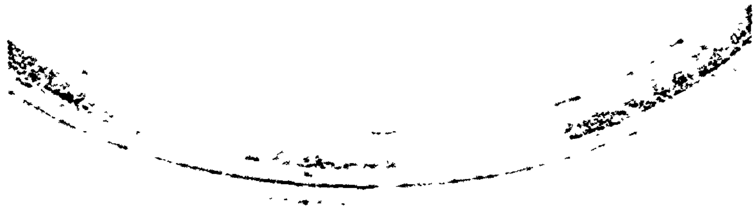
CONE NO. 2 SERIAL NUMBERED 72-4



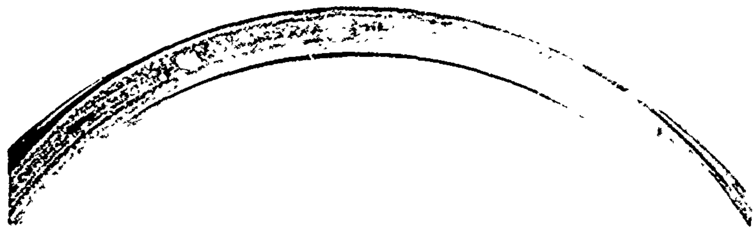
NOTE GEOMETRIC STRESS CONCENTRATION SPALL
AND DEBRIS BRUISING DAMAGE

Figure 28. Condition of Cone Raceway of SK301-10257-1 Bearing.

CONE NO. 1 SERIAL NUMBERED 72-2
(COPPER PLATED BORE & BACK FACE)



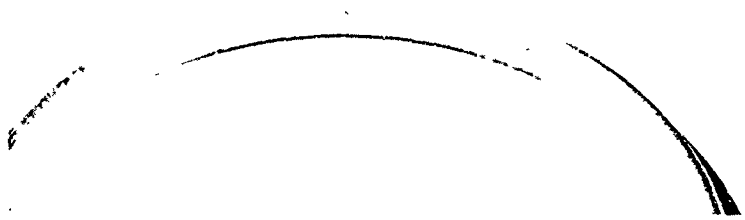
BACK FACE - FRETTED AT SUPPORT;
SCORED AT OUTSIDE EDGE



CONE BORE FRETTING CORROSION

CONE SERIAL NUMBERED 72-4 (NO COPPER PLATING)

BACK FACE HEAVILY WORN AT SUPPORT AREA



CONE BORE FRETTING CORROSION

Figure 29. Bore and Back Face of SK301-10257-1 Bearing.

CUP NO. 1 SERIAL NUMBERED 72-6

NOTE LIGHT FALSE BRINELLING PROBABLY INCURRED IN SHIPMENT AND DEBRIS BRUISING DAMAGE POSTTEST

CUP NO. 2 SERIAL NUMBERED 72-8

GEOMETRIC STRESS CONCENTRATION SPALLING AT LARGE END OF RACE IN TWO GROUPS GENERALLY CORRESPONDING TO LOCATION OF LOADING LUGS ON TEST HOUSING

Figure 30. Condition of Cup Raceways of SK301-10257-1 Bearings.

CUP NO. 1 SERIAL NUMBERED 72-6

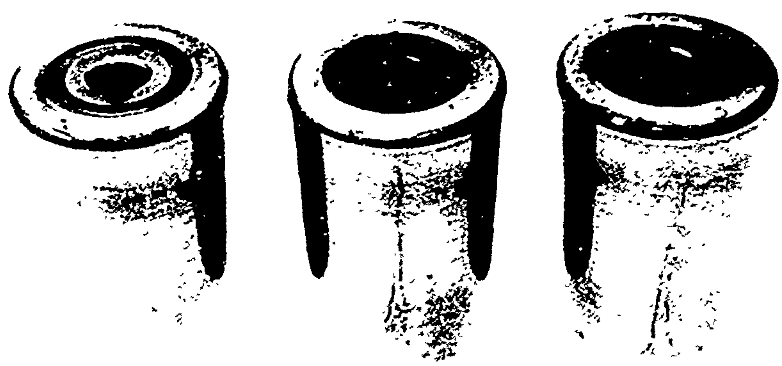
SLIGHT FRETTING CORROSION ON
O.D. (NO COPPER PLATING)

CUP NO. 2 SERIAL NUMBERED 72-8

NO SIGNIFICANT FRETTING (COPPER PLATING)

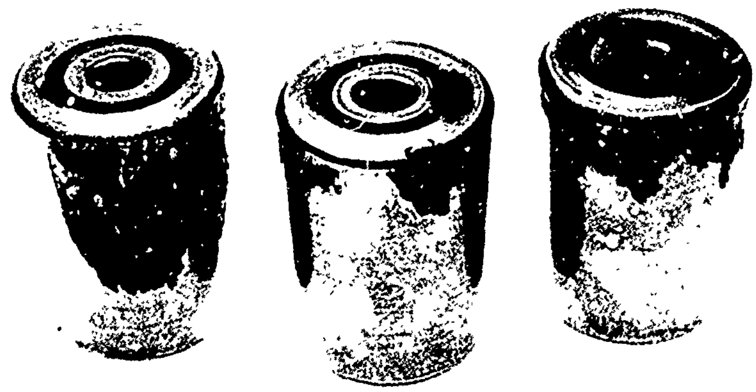
Figure 31. Outer Diameter of Cups of SK301-10257-1 Bearings.

ROLLERS FROM BEARING ASSEMBLY NO. 1



ETCH MARKS ON O.D.'S AND SPHERICAL ENDS

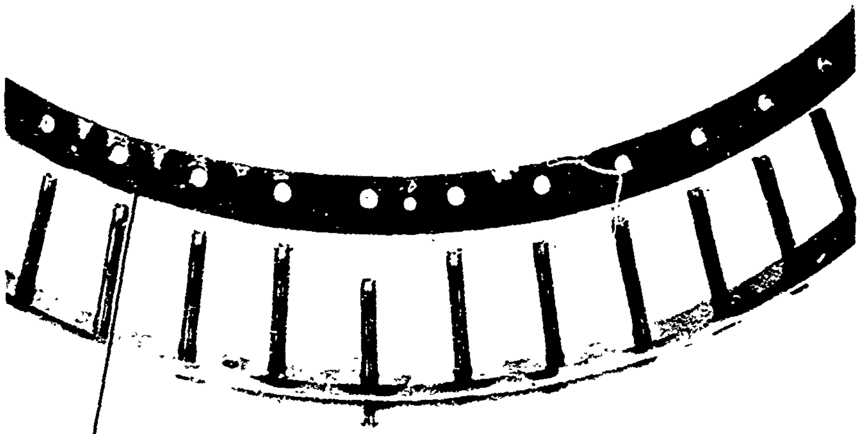
ROLLERS FROM BEARING ASSEMBLY NO. 2



GEOMETRIC STRESS CONCENTRATION FATIGUE DAMAGE

Figure 32. Rollers of SK301-10257-1 Bearings.

CAGE ASSEMBLY FROM BEARING ASSEMBLY NO. 1



NOTE HEAVY ETCHING AT ROLLER LOCATIONS

CAGE ASSEMBLY FROM BEARING ASSEMBLY NO. 2

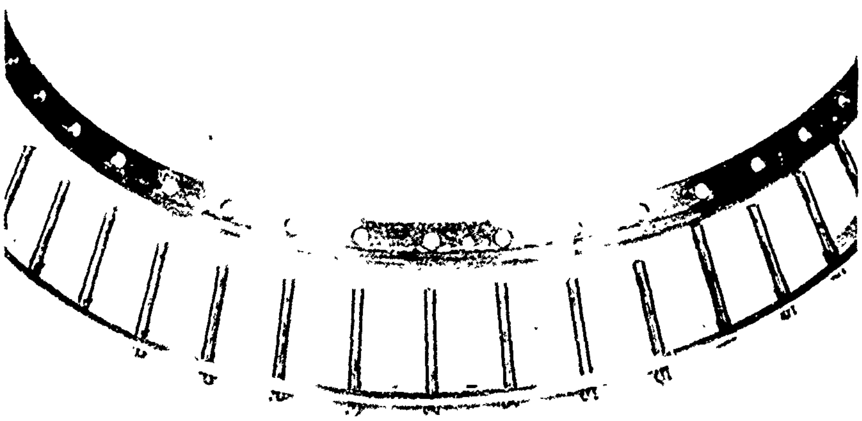


Figure 33. Cage Assembly of SK301-10257-1 Bearings.

CRACKING ON ROLLER FROM
BEARING ASSEMBLY NO. 2

NOTE THAT CRACK EXTENDS ACROSS
SPHERICAL END AND INTO PIN BORE

Figure 34. Cracked Roller.

The results of this analysis are summarized below:

Microstructure

Cone -	Case:	Fairly fine tempered martensite, some fine carbides, nil retained austenite.
	Core:	Tempered martensite, 5 percent ferrite.
Cup -	Case:	Fairly fine tempered martensite, some fine carbides, nil retained austenite.
	Core:	Tempered martensite, nil ferrite.
Roller -	Case:	Fairly fine tempered martensite, some fine carbides, nil retained austenite.
	Core:	Tempered martensite, nil ferrite.

Hardness - Rockwell "C"

	Case	Core
Cone	61	38
Cup	60	39
Roller	61	42

Ms Case Depth

	0.50% Carbon	0.80% Carbon
Cone	0.044 (0.042 Min.)	0.016 (0.005 Min.)
Cup	0.048 (0.042 Min.)	0.012 (0.005 Min.)
Roller	0.068 (0.068 Min.)	0.016 (0.005 Min.)

This metallurgical analysis showed that all components were within standard limits. Therefore, the mode of failure of the cup and rollers was of the geometric stress concentration type, and the cracking of the roller large end was directly associated with the stresses caused by the geometric stress concentration spalling.

EVALUATION OF BEARING FATIGUE LIFE

Since the actual HLH rotor shaft bearings are different from the bearings tested in this program, a relationship between test conditions and flight conditions must be defined. This relationship will establish the expected equivalent flight time on the actual bearings that has been demonstrated by the test bearings. Only the classical mode of bearing failure is included in the following discussion.

The term fatigue life ratio (d) is defined as the ratio of operating time (t) at some load level to L_{10} life at the same load level:

$$d = \frac{t}{L_{10}}$$

Since d becomes 1.0 when $t = L_{10}$, the fatigue life ratio can be considered approximately proportional to probability of failure. Thus, two bearings exhibiting the same ratio will have equal probabilities of failure. Therefore, the equivalent flight time on the aircraft bearings is obtained by equating test fatigue life ratio to the aircraft fatigue life ratio.

$$d_{Test} = d_{Aircraft}$$

$$\frac{t}{L_{10\ Test}} = \frac{\text{Equivalent Flight Time}}{L_{10\ Aircraft}}$$

$$\text{Equivalent Flight Time} = \frac{L_{10\ Aircraft}}{L_{10\ Test}} \times t$$

Table X presents the test load matrix used during this test program.

Table XI shows the test bearing L_{10} lives and the fatigue life ratio for each test bearing.

TABLE X. TEST LOAD MATRIX

Test Condition	Test Rig Loads, Lb: R_1 Actuator	Test Rig Loads, Lb: R_2 Actuator	Test Time, Hours: Bearings, Upper Assembly	Test Time, Hours: Bearings, Lower Assembly
Load 1	43700	14950	400	400
Load 2	50350	14950	690	738

The actual HLH rotor shaft bearing configuration is shown in Figure 35 . The equivalent radial loads and L_{10} lives presented in Table XII are calculated considering the mission profile and flapping angle spectrum for the HLH aircraft. Also shown in Table XII is the fatigue life ratio which

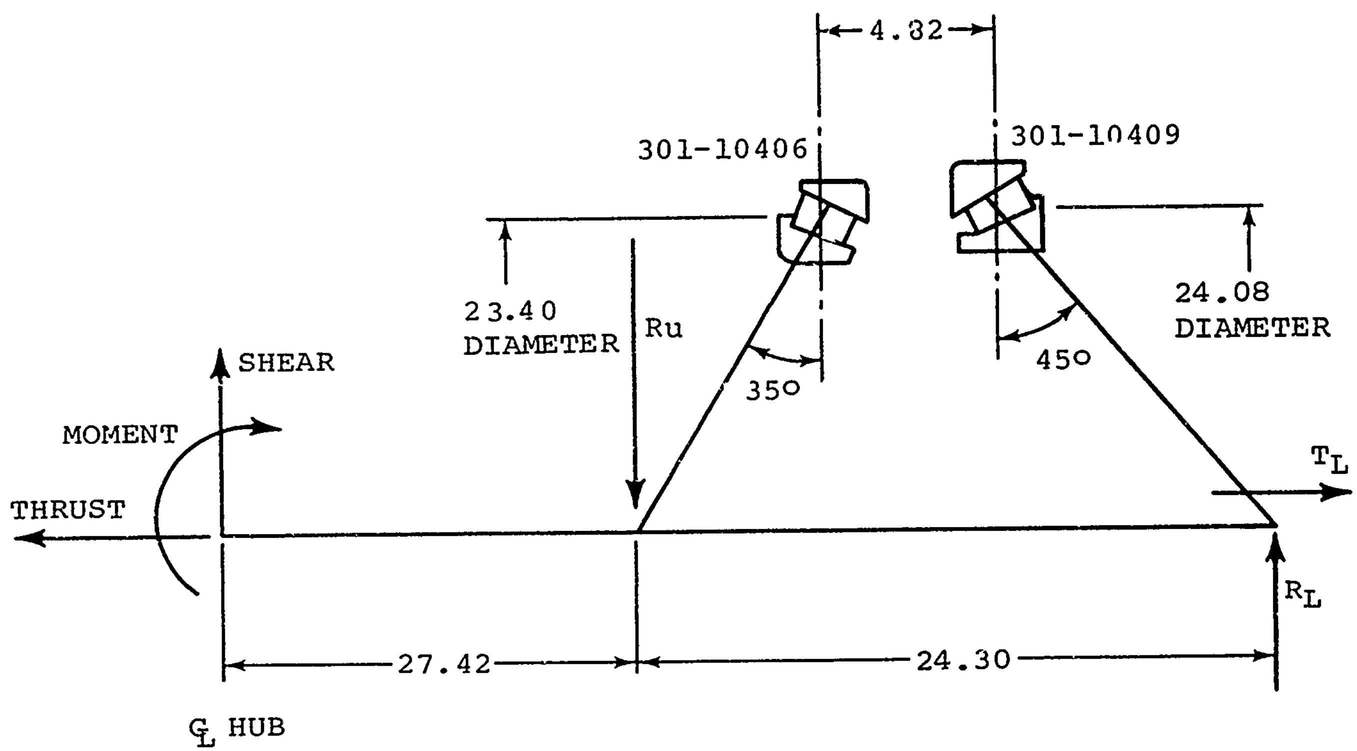


Figure 35. HLH Aft Rotor Shaft Bearing Configuration.

TABLE XI. TEST BEARING LIFE SUMMARY

Bearing Location	Bearing Part No.	Cubic Mean Equivalent Radial Test Load, lb	(1) L_{10}, hr	Test Time, hr	Ratio = $\frac{Test}{L_{10}}$	Test Results
Upper Bearing Upper Housing	SK301-10257-2	80,500	3855	1090	0.283	No Failure
Lower Bearing Upper Housing	SK301-10257-1	95,000	3407	1090	0.334	Spalled
Lower Bearing Lower Housing	SK301-10257-1	95,200	3385	1138	0.336	No Failure
Upper Bearing Lower Housing	SK301-10257-2	80,600	3840	1138	0.296	No Failure

(1) Includes a material life improvement factor of 3.0

TABLE XII. HLH ROTOR SHAFT BEARING LIFE SUMMARY

Bearing Part No.	Location	Equivalent Radial Load, lb	L_{10} * Life, hr	Fatigue Life Ratio From Test	Aircraft Flight Time Equivalent To Total Test Time, hr
301-10406	Upper Bearing	54600	4560	0.283 0.296	1290 1352
301-10409	Lower Bearing	64700	4250	0.334 0.336	1420 1428

*L_{10} Life Includes a Material Life Improvement Factor of 3.0

accrued on each of the bearings during the testing. Since this ratio corresponds to the portion of life used, multiplying the calculated L_{10} life by the ratio yields the aircraft flight time equivalent to the test time. Therefore, the testing conducted during this program has demonstrated that the actual HLH rotor shaft bearings should have an expected service life in excess of 1300 hours.

SUMMARY OF TESTING

A total test time of 1140.0 hours was accumulated on one set of HLH aft rotor shaft bearings in the lower housing assembly. The bearings in the upper housing assembly accumulated 1093.0 total hours. Of this time, 693.0 hours was at a loading condition of 70.83×10^3 pounds thrust load and a moment of 2.16×10^6 inch-pounds. The remaining 400 hours was run at a 57.50×10^3 pound thrust load and a moment of 1.96×10^6 inch-pounds.

The bearings in the lower assembly had a total test time of 1140.0 hours. Of this time, 719.5 hours was at a thrust load of 70.83×10^3 pounds and a moment of 2.16×10^6 inch-pounds. A total of 400 hours was run under a thrust load of 57.50×10^3 pounds and a moment of 1.96×10^6 inch-pounds. The initial 16 hours was run under a "no-load" condition followed by a 6.5-hour interval during which the thrust load was increased in approximately 10 percent intervals.

During the break-in period, a temperature rise of 20°F was measured under no-load conditions and 40°F under full loading conditions without preheating the oil. During testing under the prescribed operating conditions, the peak "g" levels were between 1.0-2.0. The test assembly power consumption was approximately 1.5 kw under steady-state conditions. Total lubricant flow was 3.0 gpm.

The first indication of a possible bearing failure occurred at 899.5 total hours test time on the upper assembly bearings.

Inspection of the bearings at the conclusion of the test program (1093.0 total hours on the upper assembly and 1140 total hours on the lower assembly) revealed that the lower bearing in the upper assembly had failed. The fatigue failure (spall) may have been influenced by the numerous failures of the preload bolts and edge loading of the rollers caused by the lateral growth of the bearing cone when the preload plate deflected. The failed bearing had 38 spalled rollers and the cup had several spalls in the area of the highest load application and greatest housing stiffness.

The tungsten-carbide bearing seats exhibited excellent wear properties in spite of the severe conditions imposed on them

during the test program. From the experience of many assemblies and disassemblies, it appeared that the hard (Rc-72) and smooth surface (3-12 in. rms) of the tungsten-carbide material is very beneficial for installing and removing the bearings from the shaft, as was the inclusion of an oil groove in the bearing seats. The use of 440C stainless steel deposited by a spray welding process for bearing seats does not appear suitable for this application.

Posttest visual hardware inspection indicated that the alkaline copper plating on the bearing cone bores, combined with an increase in interference fits, contributed to improved surface conditions at the bearing and shaft interface.

Of the analytical and experimental methods investigated to determine bearing preload spacer lengths, the experimental method was more successful in that it was easier, more reliable, and required fewer regrinding operations to establish the desired preload setting.

CONCLUSIONS

1. The use of large-diameter, steep-contact-angle tapered roller bearings to support the HLH rotor shaft provides good performance and adequate fatigue life.

2. Azimuth variation in stiffness of the bearing support housing does affect the internal load distribution of the bearing. Initiation of spall failures is associated with the mounting arms of the housing which produce local concentrated peak loads.

3. Various preloads from line-to-line to 0.0015 inch appear to be adequate and practical for the HLH rotor shaft bearings. This preload range is sufficient to provide good performance without excessive heat generation.

4. Low oil film thickness does not appear to affect bearing performance or fatigue life. There was no evidence of surface distress on the raceways and roller spherical end surfaces.

5. The pin-type cage used in the HLH rotor shaft bearings is adequate and allows maximum bearing capacity. There was no evidence of cage distress or pin wear at completion of testing.

6. Copper plating the cup outer diameter, cone bore, and cone back face appeared to minimize fretting damage during operation.

7. The type of preload clamp-up plate used in this test program had a significant effect on reducing bearing performance and life.

8. Based upon the results obtained from this test program, it is expected that the actual HLH aft rotor shaft bearing performance should be adequate and a life of more than 1300 hours should be achieved.

RECOMMENDATIONS

Although the data generated in this test program is conclusive as far as the design feasibility of using large-diameter, high-contact-angle tapered roller bearings, additional testing in the actual HLH transmission will be required to determine the performance of the bearing under actual loading. The effects of housing and shaft deflections on bearing internal loading will have to be evaluated by testing. If evidence of edge type loading occurs on the bearing races and roller large end surfaces, additional internal modifications (such as crown races and increased roller crown) may be required to minimize this effect. It is not recommended that these changes be made until the rotor shaft bearings are evaluated in the actual HLH transmission environment.